DESIGN OF THE FLOW DISTRIBUTION SYSTEM FOR A PERIODIC COOLING TOWER

Layne Walter Good

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by

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ABSTRACT

The periodic cooling tower consists of a series of rotating metal discs, which are partially submerged in the cooling water that is circulated from the condenser of a steam driven electric power plant. The surface of the water between the discs is covered by oil to eliminate water losses to the atmosphere as the discs are cooled by forced air convection. Poor flow distribution of the circulating water to the discs can adversely effect the waste heat rejection capabilities of the periodic cooling tower. This thesis develops methods for predicting the pressures developed in various portions of a periodic cooling tower module. That makes it possible to design a flow distribution system which will deliver a uniform quantity of circulating water to the disc spaces along the entire disc matrix.

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LIST OF SYMBOLS USED IN TEXT

Symbol	Definition
A_o	orifice area
A_R	trough cross-sectional area
A_T	cross-sectional area of immersed gap between plates
c_d	discharge coefficient
D_e	hydraulic diameter
d_o	orifice diameter
D_T	basin depth
e	statistical measure of surface roughness
f	friction factor
F_D	drag force
g	acceleration of gravity
H_o	head above orifice center
h_d	head developed by rotating discs
h_f	frictional head loss along trough
h_o	head developed by orifice restriction
h_v	velocity head
H_w	head on weir
ℓ	trough length
L	waterline length of partially submerged discs
L_c	weir crest length
L_T	trough length from end to point of interest

Symbol	Description
n	number of orifices per foot of trough
N	number of gallons of flow per minute per foot of trough
P	wetted perimeter of trough
Q	volume flow rate
R	disc radius
r_{avg}	average radius of the submerged portion of discs
Re	Reynolds number
s	disc spacing
V	water velocity
V_p	plate velocity
V_x	local water velocity in the x-direction
$\dot{W}$	mass flow rate
$\dot{W}_f$	mass of water removed laterally per foot of inlet trough
W_T	trough width at waterline
Y_{avg}	average of the distance from the disc center to the point of interest and Y_s
Y_e	depth of equivalent flat plate
Y_s	distance from the disc center to the waterline

Greek Symbols

ΔP	change in pressure
ΔP_s	head developed by spacer drag force
ν	kinematic viscosity
ρ	density of water

Symbol	Description
τ	shear force
ω	disc rotational speed

INTRODUCTION

Steam driven electric power plants require that approximately six units of energy be rejected as waste heat in order to produce four units of electrical energy. This waste heat can be removed from the turbine condenser in a number of ways.

Plants located near oceans, lakes, or rivers have used these large bodies of water for once through cooling of their condensers. This method involves circulating large quantities of cooling water through the condenser once and then discharging the heated effluent back into the environement. The resulting increase in the water temperature near the outlet, due to the rejection of waste heat, can upset the ecological balance in the vicinity of the discharge line. Another disadvantage of this method of cooling is that not all proposed plant sites have a large body of water in close proximity for easy, economical use.

Wet cooling towers splash the tower circulating water down a wooden lattice, after it has been heated by passing through the condenser. As the water falls it is cooled partially by either forced or natural convection air flow. However, about eighty percent of the heat transfer that takes place is by evaporation.[1] The moist air leaving the cooling tower may form a fog plume that can obscure visibility. More importantly a one thousand Megawatt plant can loose as much as 9.3 million gallons of water per day to the surrounding atmosphere[2].

Such large quantities of water may be unavailable or prohibitively expensive.

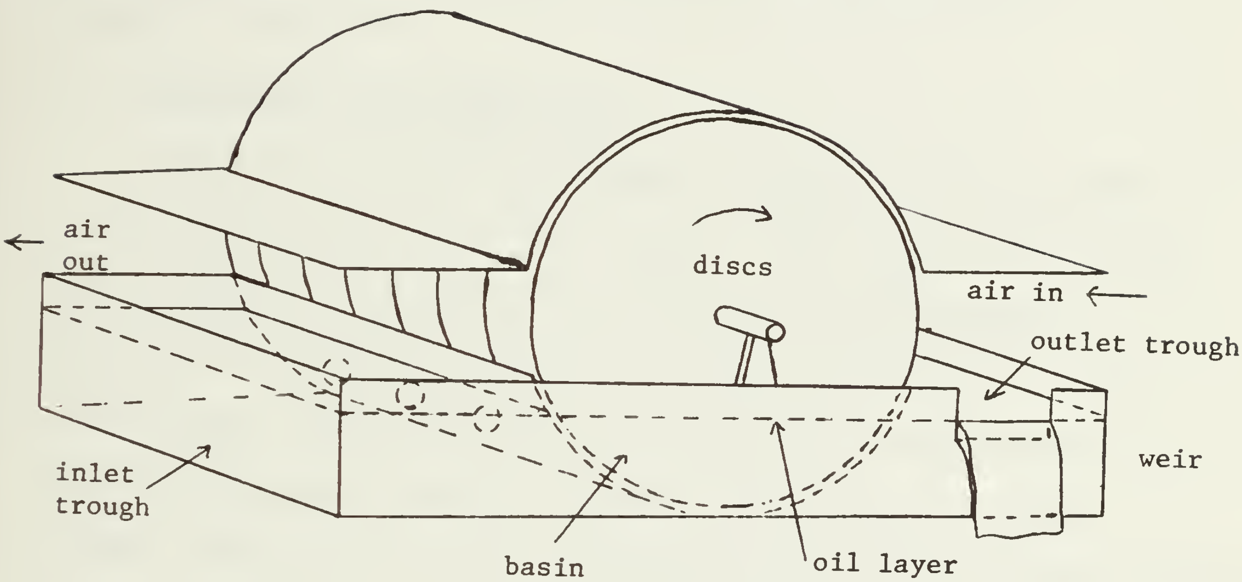
In dry cooling towers, the circulating water is alternatively heated in the condenser and cooled as it passes through air cooled finned tubes. There are not water losses due to evaporation, but because of the poor heat transport properties of air, a much larger surface area is needed to reject the same amount of heat as a wet cooling tower. The increased surface area coupled with its greater expense means a capital cost of three to four times more for a dry cooling tower than a wet cooling tower.[2]

The periodic cooling tower consists of a series of rotating metal discs, which are partially submerged in the water that is circulated from the condenser for cooling. Figure I shows a conceptual design for one module of a periodic cooling tower. The hot circulating water from the condenser is pumped into the inlet trough. From there it is transmitted through orifices to the rotating partially submerged discs. The surface of the water between the discs is covered by oil, whose properties have been selected to wet the plates, as well as resist emulsification and evaporation. The oil layer is heated by the entering water and as the rotating discs emerge from the bath they bring a portion of the oil with them. The discs, which absorb the majority of the heat from the circulating water, and the oil are then cooled by forced convection air flow between the disc spacings. Some of the cooled oil drains back onto the oil layer, while the rest is forced from the discs as they re-enter

the water, by the buoyant forces on the oil. The process is repeated and the heat from the circulating water is rejected into the atmosphere. The cooled water then enters the outlet trough and is discharged by Weir into the circulating water going back to the condenser. The oil layer is contained in the basin by baffles.

The five main advantages of the periodic cooling tower are:

1. The circulating water is normally covered by a layer of oil to eliminate water losses to the atmosphere.
2. The capital cost of a periodic cooling tower is less than a conventional dry cooling tower with the same waste heat rejection performance.
3. The periodic cooling tower can be operated wet (i.e. the oil cover removed) to eliminate the large generating capacity losses of conventional dry cooling towers during hot weather.
4. Rejection of heat to the atmosphere reduces the ecological impact of waste thermal energy to a minimum.
5. The oil layer prevents corrosion and because of the periodic flow reversals there are no regions of flow stagnation. This helps the disc surfaces stay clean.



Conceptual Design of a Periodic Cooling Tower Module

Figure I-1

CHAPTER 1

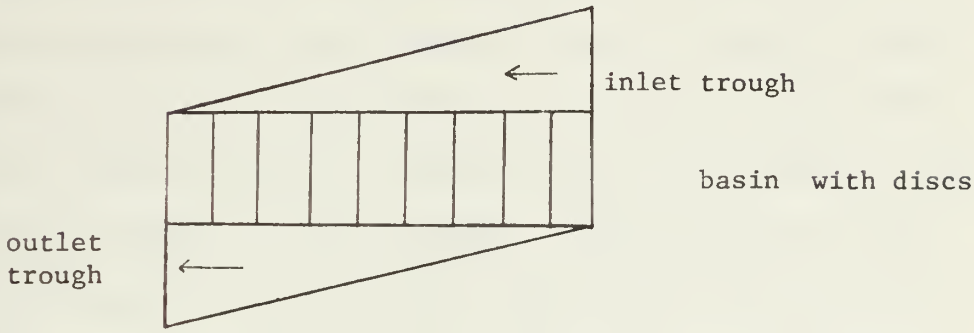
THE NEED FOR EVEN FLOW DISTRIBUTION

The design of the flow distribution system must be conservative enought to ensure that there are no irregularities in the flow of circulating water through the discs. Poor flow distribution can adversely effect the waste heat rejection capabilities of the periodic cooling tower.

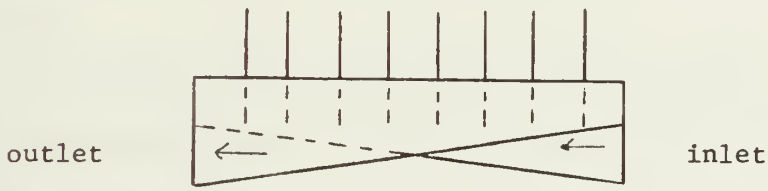
1.1 Possible Trough Configurations

The flow rate of circulating water to each disc in the periodic cooling tower module has to be a constant. This helps to ensure that all portions of the disc matrix are rejecting an equal amount of waste heat. A uniform distribution of water to the discs, from the inlet trough, can be maintained by reducing the cross-sectional area in proportion to quantity of water lost laterally along the trough. Some possible trough configurations are shown in Figure 1.1-1a,b,c. Those designs are not as compact, nor as simple as that shown in Figure 1.1-1d. If that trough is wide enough, then it will approximate a reservoir with a constant pressure head along the full length of the trough. The discs will then receive an even distribution of circulating water, even though the cross-sectional area of the trough is maintained constant along its length.

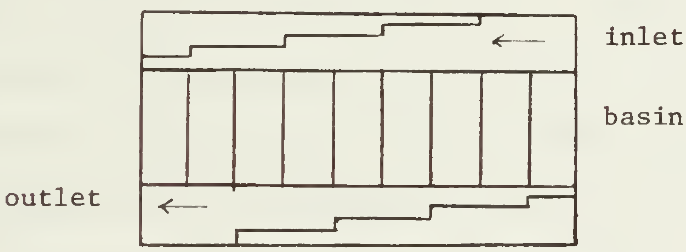
a) Reducing the cross-sectional area by decreasing the trough width (top view)



b) Reducing the cross-sectional area by decreasing the trough depth (side view)



c) Reducting the cross-sectional area with low density inserts (top view)



d) Troughs with constant cross-sectional area (top view)

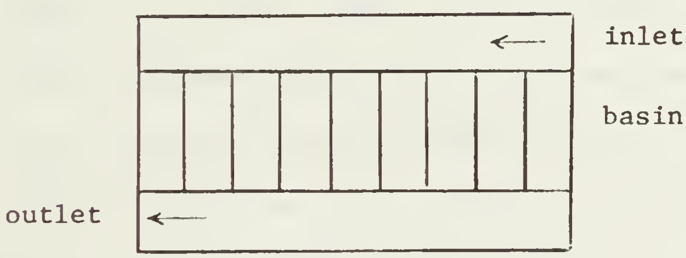


Figure 1.1-1: Possible Trough Configurations

1.2 Source of Flow Irregulatities

The loss of mass along the trough to the discs (for a constant cross-sectional area trough) becomes important when the trough is narrow. At the beginning of the inlet trough there is, of course, a larger head than at the end. Because there is a high flow rate at the entrance, there is a high pressure gradient at that location. This means that a curve of pressure head versus trough length will show a steep slope near the entrance. Near the end there is a lower flow rate, a smaller pressure gradient and therefore a shallower slope in the curve. This is illustrated in Figure 1.2-1.

At the beginning of the outlet trough there is again a larger pressure head than at the end. However, in this case the flow rate through the cross-sectional area of the trough starts out small and gets larger as more mass is added along the trough. So in this case the pressure gradient is small at first and gets progressively larger. This means that, for the outlet trough, the pressure head decreases more rapidly the further the flow progresses down the trough. This is illustrated in Figure 1.2-2.

The possibility of flow irregularities increases as the ratio of the maximum head to the minimum head gets much greater than one. Figure 1.2-3 illustrates such a case. Therefore the narrower the inlet and outlet troughs, the larger the pressure drops along the trough and the greater the maximum head across the module must be. A large maximum head will make the pressure drops along the troughs insignificant and there will be an even distribution of flow to the disc spaces. However, if the troughs are made wide enough (in the limiting case a pair of reservoirs) then the

Figure 1.2-1 Inlet Trough Flow Rates and Pressure Heads

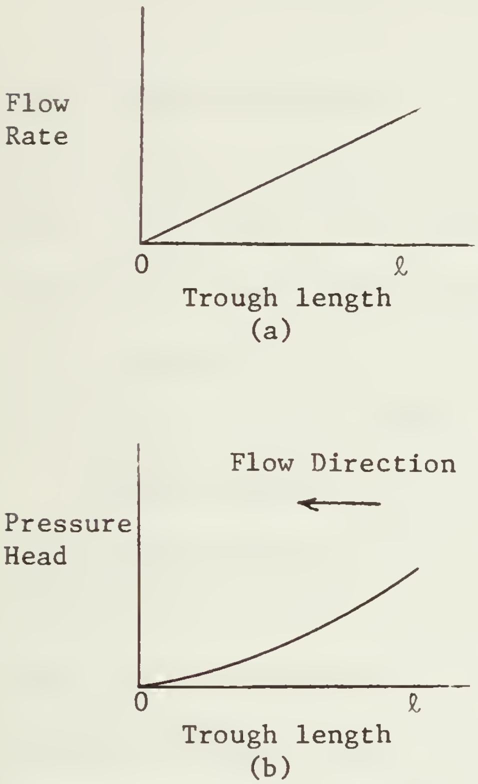
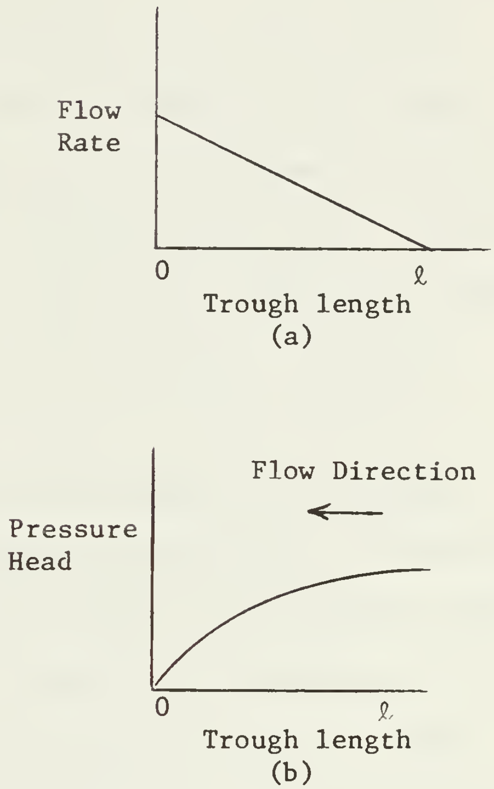


Figure 1.2-2 Outlet Trough Flow Rates and Pressure Heads



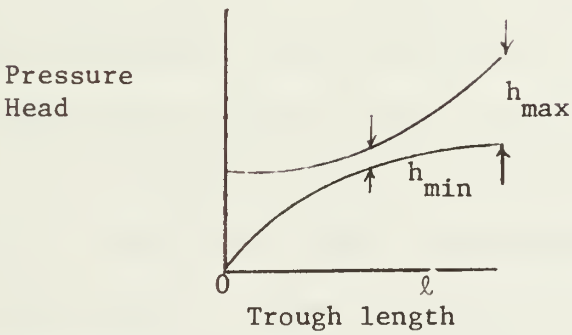


Figure 1.2-3 A Case Where Flow Irregularities May Exist

maximum head can be much smaller and there still will be a uniform distribution of flow between the discs.

1.3 The Need to Minimize Trough Size

The proposed erection scheme is to mount the periodic cooling tower modules vertidally above each other (see Figure 1.3-1). This stacking permits a denser cooling tower, greater arrangement flexibility, shorter piping runs, and optimum use of existing exhaust fans. By decreasing the trough widths the amount of water to be supported as well as the structural weight can be significantly reduced. That would in turn reduce capital costs.

Before a reduction in trough width can be made, it is necessary to know what effects it will have on the flow distribution. Conservative predictions of the head losses along the troughs and the pressure drop across the rotating discs are needed to assist in the assessment of a particular design. There is the additional need to develop a simple yet effective means of controlling the water level in the module so that the oil layer remains confined to the basin. Recent designs of the periodic cooling tower have included discs which have hollow centers, and large fluctuations in the water level are undesirable. Departures from the optimum setting will result in lowering the heat rejection efficiency of the periodic cooling tower, by either not utilizing all the disc surface in the circulating water or reducing the air side heat transfer area when the water level is too high. The subsequent chapters will deal with each of these areas for concern and yield a method for designing the flow distribution system of a periodic cooling tower.

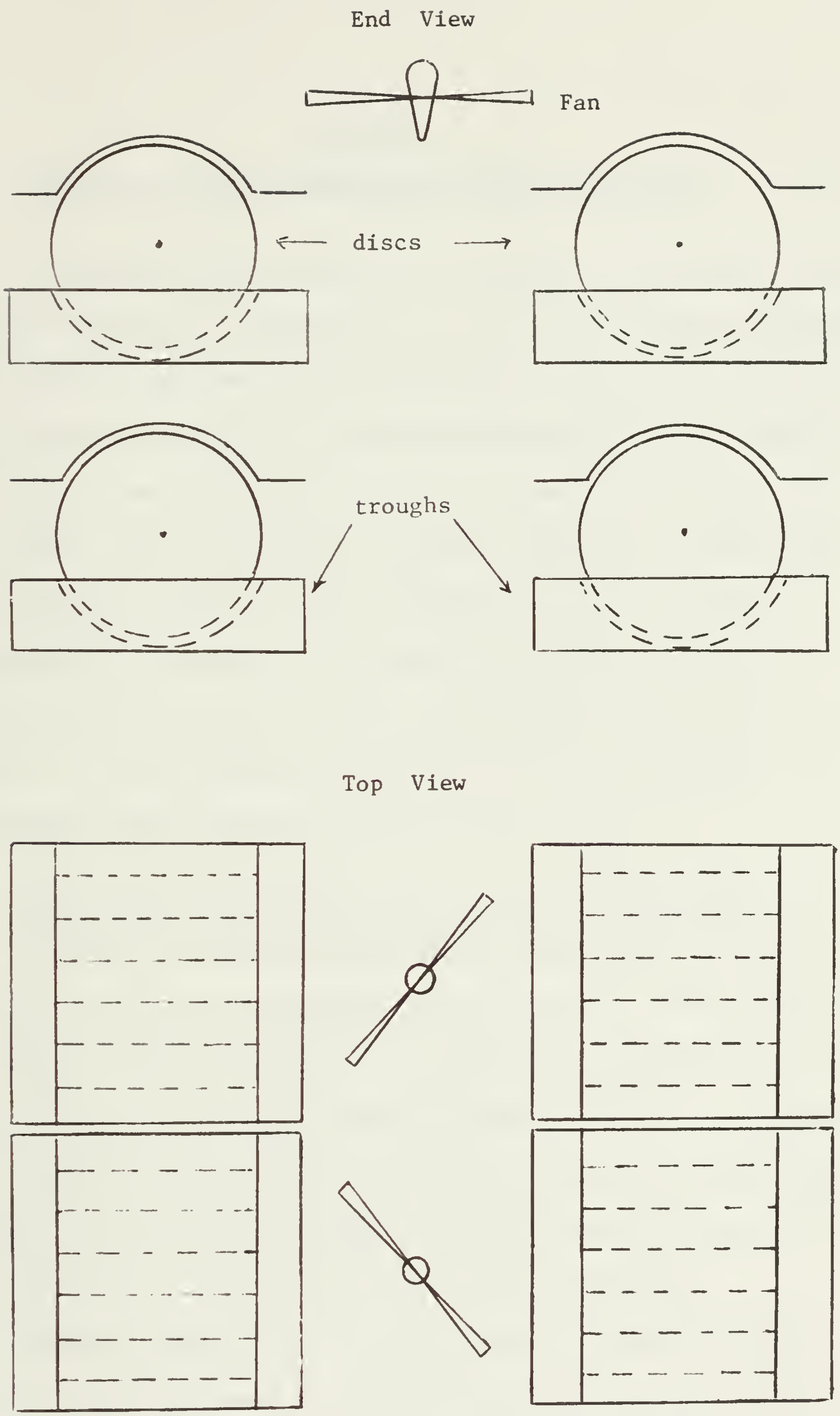


Figure 1.3-1 Proposed Module Erection Scheme

CHAPTER 2

PREDICTION OF THE PRESSURE DROP ACROSS THE DISCS

Experimental measurements of the pressure drop across rotating discs of one foot in diameter and five feet in diameter were taken. The results of the experiments were then compared to the pressure drop values predicted by two simple mathematical models. The most conservative model was then selected to make future estimates of the pressure head developed by partially submerged rotating discs.

Separate tests were run to demonstrate that there was no measureable difference in the pressure head developed by the rotating discs, whether water was being pumped through the basin or if the discs were rotating in a stagnant trough. All experiments were conducted without the oil layer.

2.1 Experimental Results with Periodic Cooling Tower Mockups

A series of periodic cooling tower mockups have been constructed to assist in optimization of the design. Such things as oil type, operating temperatures, air turbulence stimulation, optimum water levels, disc rotation speeds and water distribution methods have been examined with them[1,2,3]. Two of the mockups were used in the flow distribution studies. One had discs five feet in diameter and the other had discs one foot in diameter.

An illustration of the one foot diameter model is shown in Figure 2.1-1. The speed of rotation was varied by positioning

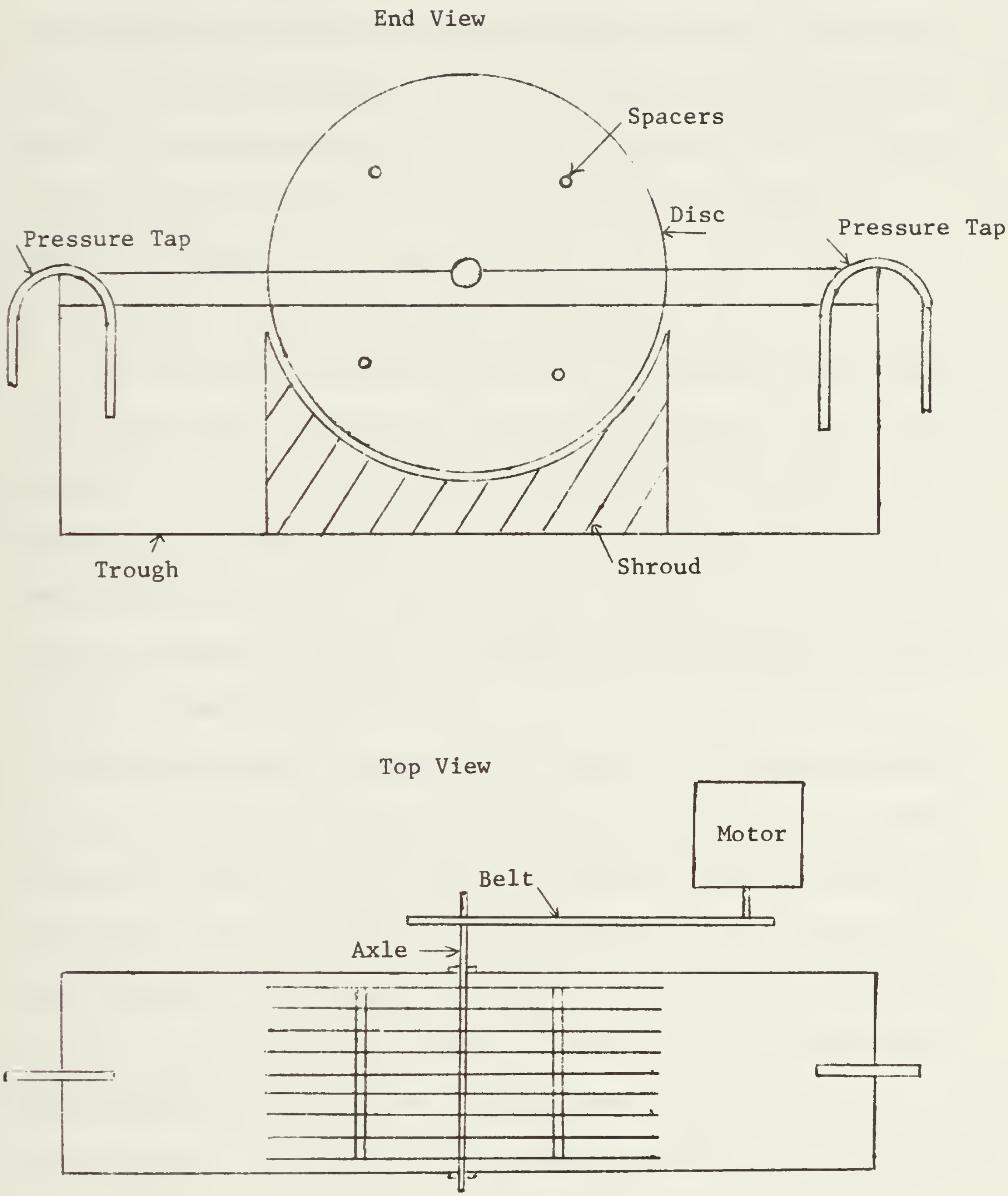


Figure 2.1-1 Experimental Mockup with 1 Foot Diameter Discs

a rubber belt on the variable diameter pulleys located on the disc axle and the electric motor. The lower portion of the discs were shrouded to minimize backflow and loss of pressure head. The change in pressure due to the rotation of the discs was measured by a manometer inclined to an angle of 12.4°. The pressure taps were located in the stagnant portions of the trough.

The disc matrix consisted of twelve .03 inch thick sheet metal discs spaced every .25 inches. The discs were mounted on a .25 inch axle which rotated on roller bearings mounted on the trough. The trough itself was made of .5 inch plexiglass and had external dimensions of 4.5 in. x 7.5 in. x 22 in. There was a .063 inch clearance between the shroud and the discs. The pressure taps had an inside diameter of .19 inches.

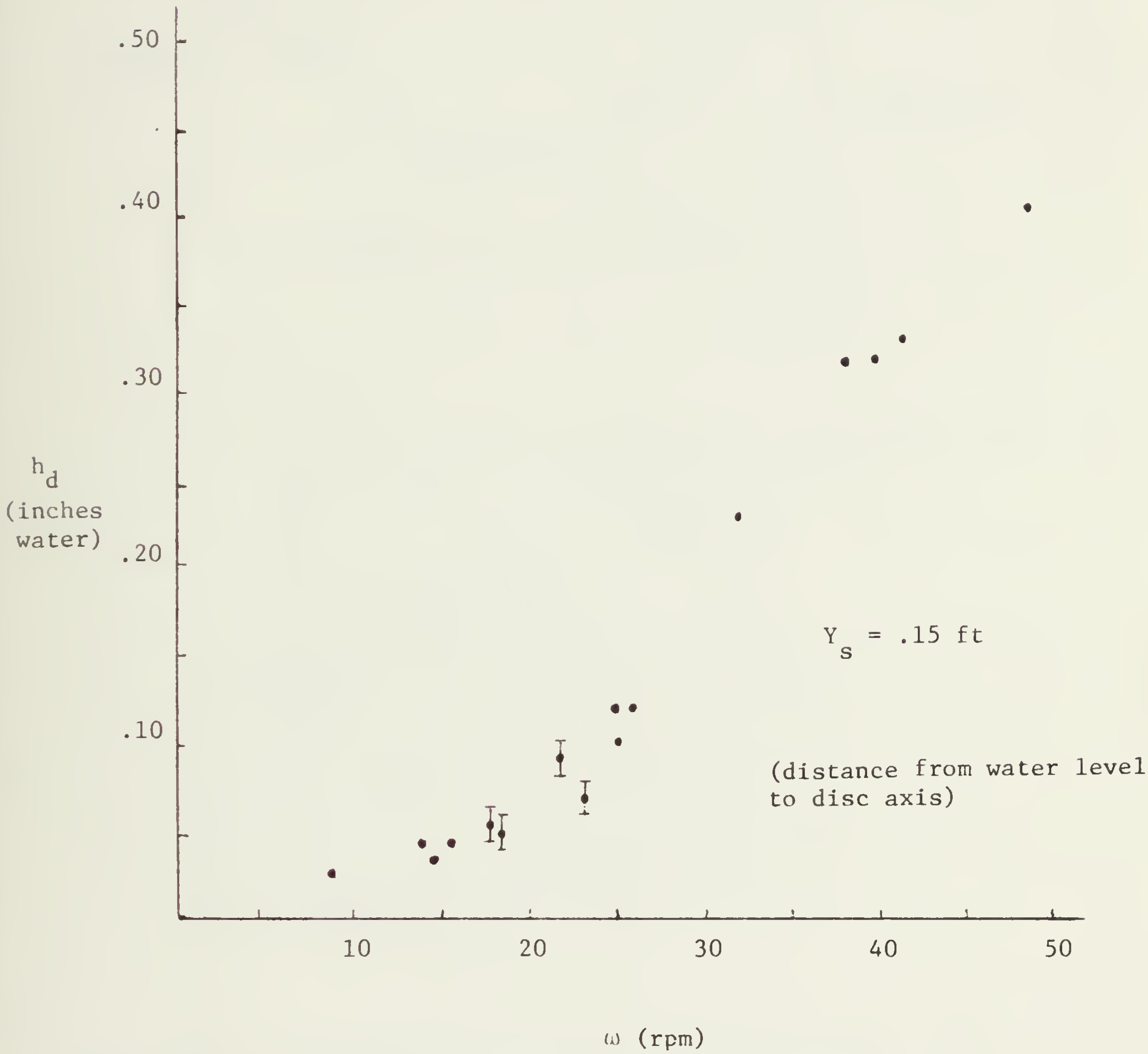
This mockup didn't prove to be as useful as was hoped. Ideally the results from such a small model could be scaled up to estimate the head developed by a full scale rig. However, the discs were separated by spacers, which succeeded in keeping them rigidly in place, allowing for close clearances. Unfortunately the spacers also cut through the water. The drag forces they created made an additional contribution to the pressure head developed by the rotating discs. The results of experiments carried out on the one foot diameter mockup helped to validate the simple mathematical models that were developed, but not without the added problem of estimating the pressure head developed by the spacers. In addition

there was no suitable way to pump water through the basin to see if that had any effect on the pressure drop. For those reasons another set of experiments were conducted on the larger mockup. The experimentally determined values for the smaller model are shown in Figure 2.1-2.

An illustration of the full scale mockup is shown in Figure 2.1-3. The discs were rotated by an electric motor through a hydraulic transmission and belted pulleys. Rotational speed was adjusted by a variable resistance coil. Again the lower portion of the wheel was shrouded. This time the gap was 0.3 inch. The ratio of gap to disc radius was of the same order as that in the smaller model so that head losses due to backflow should again be minimal. Pressure readings were taken with the same inclined manometer. Because the shroud extended above the waterline the taps could not be placed in the stagnant region of the trough to measure the static head across the discs. Instead holes were drilled in the shroud below the waterline and the taps mounted perpendicular to the shroud and flush with it. They were held rigidily in place with modeling clay.

The disc matrix consisted of eight, five feet in diameter, 22 gauge sheet metal discs spaced every 1.13 inches. The discs were mounted on a 4.5 inch hollow axle which rotated on roller bearings. the trough itself consisted of a three sided wooded box with a .5 inch plexiglass front, and had the external dimensions of 1 ft x 3 ft x 5.8 ft.

Figure 2.1-2 Experimental Results of the One Foot Diameter Discs, Showing the Relation of Rotational Speed to Pressure Head Developed



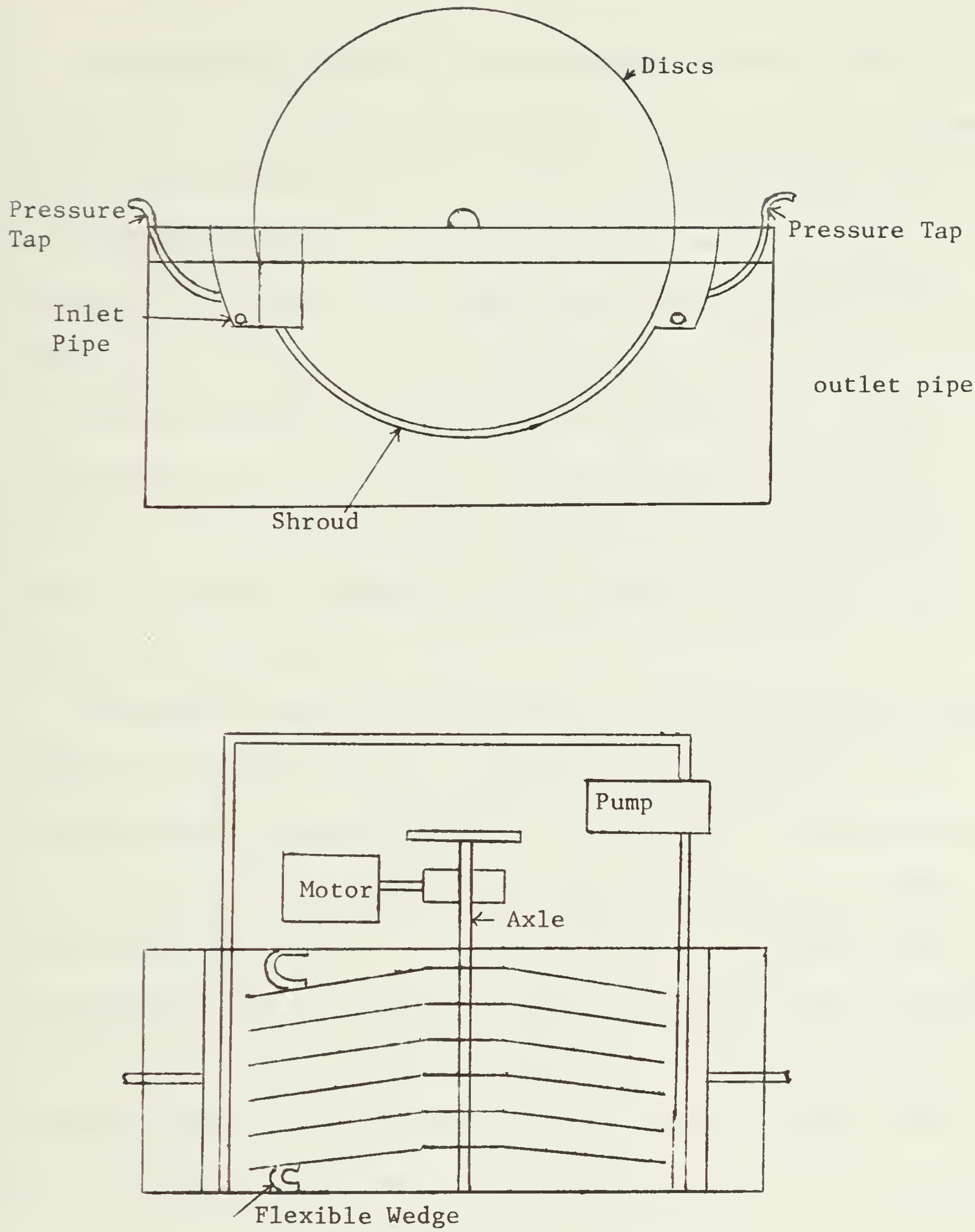


Figure 2.1-3 Experimental Mockup with Five Foot Diameter Discs

The discs were dished to improved rigidity without adding spacers, but this created some distortion in them and also created large clearances between the end discs and the trough ends. As the discs rotated these were large variable clearances. To eliminate head losses through these gaps, flexible wedges of foam rubber were installed as shown in Figure 2.1-3.

The experimental results are shown in Figure 2.1-4. These results helped to confirm that a relatively simple mathematical model can conservatively predict the pressure head generated by a matrix of partially submerged rotating discs, in an basin with no flow through it.

The maximum flow rate envisioned for a periodic cooling tower is 25 GPM per footof discs. To demonstrate experimentally that this would have a negligible effect on the pressure head developed by the rotating discs, a small centrifugal pump was used to pump water across the basin. The flow rate was adjusted until a flow meter in series with the pump and trough indicated a steady flow rate of 23 GPM per foot. There proved to be no significant difference between the case when flow was present and when it was not. The results are shown in Figure 2.1-4.

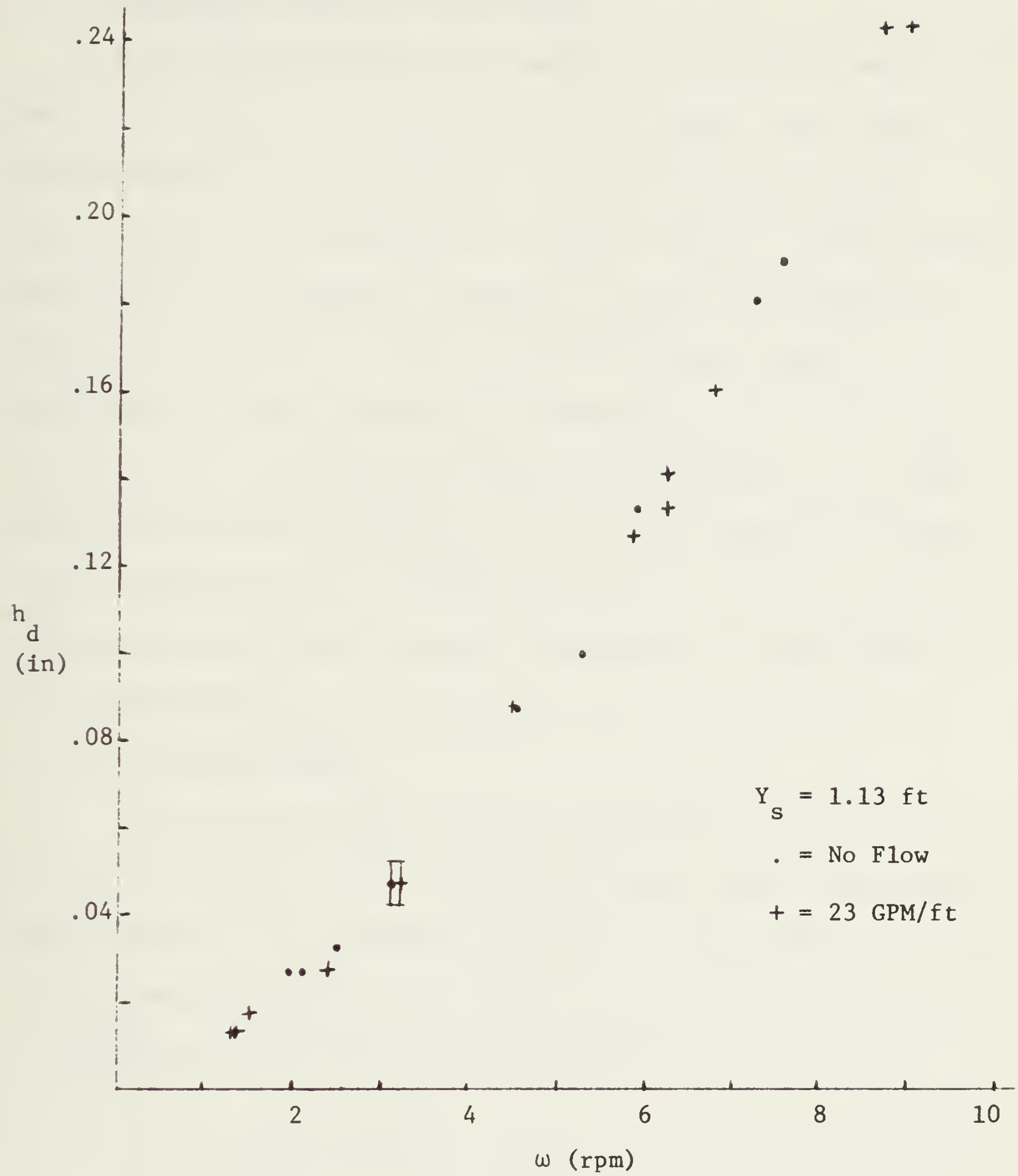


Figure 2.1-4 Comparison of Pressure Drop Across the Basin Both With and Without Flow Across It (5 ft diameter discs)

2.2 Mathematical Models and Application

A large series of experiments measuring the pressure head developed across the rotating discs could be made. Such things as disc diameter and disc spacing, as well as rotational speed and the amount of disc submergence could be varied. A more economical method would be to develop a mathematical model to conservatively predict the head. Then as the cooling tower design progresses and dimensions and other parameters are changed the effect on the head will be known. Two simple models were developed and, based on the experiments outlined in Section 2.1., the most conservative one was chosen for further use. Both models assumed that the flow of circulating water through the inner trough made no contribution to the head developed by the rotating discs.

a) Flat Plate Model

The simplest model would be to reduce the rotating discs to a series of parallel flat plates of equivalent area, moving with a linear velocity of the same magnitude as the average angular velocity of the submerged portion of the disc. With those simplifications the change in pressure across the discs can be calculated by[4]:

$$\Delta P = 4f \left(\frac{L}{D_e}\right) \left(\rho \frac{V_p^{\,2}}{2}\right) \qquad (2.2.1)$$

where f is the friction factor. For galvanized steel in turbulent flow it is a function of $.0005/D_e$ and can be found in Moody's Graph of friction factor versus Reynolds numbers published in a number of

sources[4]. Also

$$f = \frac{24}{Re} \quad \text{(Laminar flow)} \tag{2.2.2}$$

The hydraulic diameter is defined as:

$$D_e = \frac{4(\text{flow area})}{(\text{wetted permimeter})} = \frac{4\ S\ Y_e}{(2\ Y_e + S)} \tag{2.2.3}$$

and the Reynolds number is

$$Re = \frac{V_p D_e}{\nu} \tag{2.2.4}$$

The disc spacing is S, Y_e is the depth of the equivalent flat plate (derived in Appendix A and shown in Figure A-1), L is the waterline length and V_p is the plate velocity.

The plate velocity may be found by:

$$V_p = 2\pi r_{avg}\omega \qquad (\omega \text{ in rph}) \tag{2.2.5}$$

and

$$r_{avg} = \frac{Y_s + R}{2} \tag{2.2.6}$$

where ω is the rotational speed of the discs, Y_s the distance from disc center to the waterline and R is the disc radius.

For the case of turbulent flow equation (2.2.1) becomes (ΔP in inches of water):

$$\Delta P = 9.08 \times 10^{-9}\ fL\rho \left(\frac{2Y_e + S}{SY_e}\right) (r_{avg}\ \omega)^2 \tag{2.2.7}$$

and for laminar flow:

$$\Delta P = 8.69 \times 10^{-9}\ L\rho\nu \left(\frac{2Y_e+S}{SY_e}\right)^2 r_{avg}\omega \tag{2.2.8}$$

Table 2.2-1 shows a summary of the disc matrix dimensions and water conditions for both mockups tested. Using that information and either equation (2.2.7) or equation (2.2.8), the pressure head developed by the rotating discs can be calculated. The transition to turbulent flow is at Re $\simeq$ 2300.

The disc spacers in the small mockup cut through the water as the discs rotated. Because of their drag forces an additional pressure head was developed (ΔP_s), which must be added to the pressure head generated by just the rotating discs. The derivation of ΔP_s is shown in Appendix B and yields:

$$\Delta P_s = 2.78 \times 10^{-5}\ \omega^2 \ (\omega \text{ in rpm}) \qquad \text{(B-1)}$$

Tables 2.2-2 and 2.2-3 show the predicted pressure heads. These predicted heads are then compared to those found experimentally in Figures 2.2-1 and 2.2-2. As can be seen in the figures, the flat plate mathematical model overpredicts. This would lead to an overestimation of the beneficial effects of the rotating discs on helping to curb possible flow irregularities. Therefore a more conservative and realistic model is needed.

TABLE 2.2-1

DISC MATRIX DIMENSIONS AND

WATER CONDITIONS DURING EXPERIMENTS

Disc Diameter (ft)*	1.0	5.0
S	.02	.09
Y_c	.26	.98
D_e	.04	.18
Y_s	.15	1.13
L	.95	4.46
r_{avg}	.33	1.82
$\frac{.0005}{D_e}$ ft^{-1}	.013	.003
ρ lb_m/ft^3	62.2	62.3
ν ft^2/hr	0.36	.055

*All lengths in feet.

TABLE 2.2-2

FLAT PLATE MODEL PREDICTIONS OF PRESSURE HEADS ACROSS ROTATING ONE FOOT DIAMETER DISCS

ω(rpm)	Re	f	h_f	ΔP_s	$h_d + \Delta P_s$ (in)
5	698	-	.018	.001	.019
10	1395	-	.036	.003	.039
15	2093	-	.054	.006	.060
20	2792	.0135	.112	.011	.123
30	4188	.0128	.239	.025	.264
40	5584	.0122	.407	.044	.451
50	6980	.0120	.625	.070	.695

TABLE 2.2-3

FLAT PLATE MODEL PREDICTIONS OF PRESSURE HEADS ACROSS ROTATING FIVE FOOT DIAMETER DISCS

ω(rpm)	Re	f	h_d (in)
1	2238	-	.007
3	6714	.0096	.058
5	11190	.0087	.146
6	13428	.0085	.205
7	15666	.0082	.270
9	20142	.0077	.418
10	22380	.0075	.503

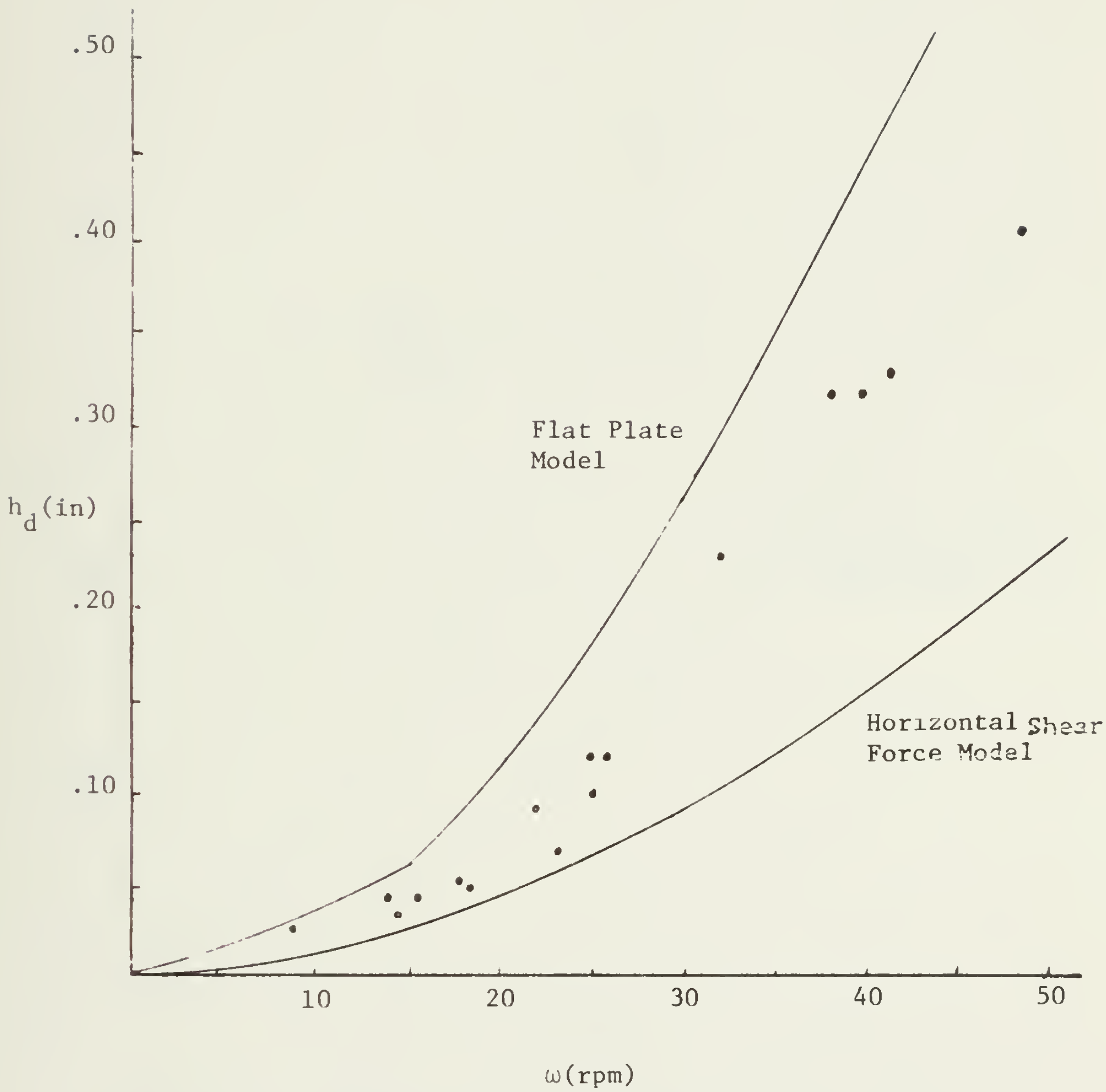


Figure 2.2-1 The Flat Plate and Shearing Model Predictions Compared to the Experimental Results (1 Foot Diameter Discs)

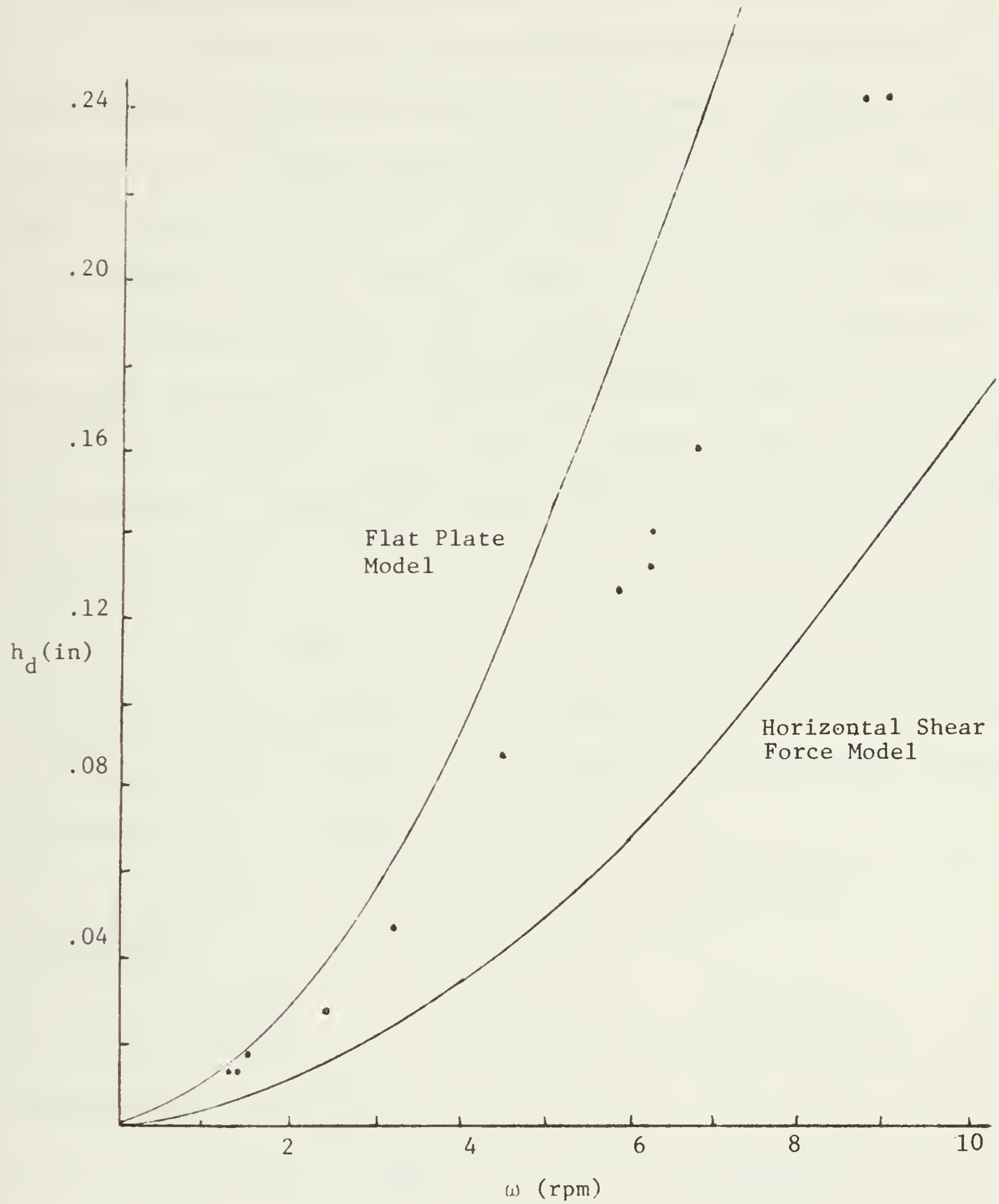


Figure 2.2-2 The Flat Plate and Shearing Model Predictions Compared to the Experimental Results (5 foot Diameter Discs)

b) Horizontal Shear Force Model

An alternative model has been proposed that is based on the local horizontal shear force between the water and the rotating discs. The disc rotation induces flow patterns in the water bath[5]. It is assumed, to aid in developing a simple model, that the fluid momentum, flow variations in the z direction, normal to the discs, and the shear forces in the y, vertical direction, can be ignored. It is also assumed that the change in pressure as x, horizontal direction parallel to the discs, varies is not a function of y. Then only the horizontal shear force, which is proportional to the relative speed between the disc and the water flow, is considered and[4]:

$$\tau_x = \frac{1}{2} f \rho(\omega y - V_x)^2 \qquad (2.2.9)$$

where V_x is the local water velocity in the x direction and y is the distance to that locality from the disc center. Because the disc radius is much larger than the disc spacing then

$$D_e = 2 S \qquad (2.2.10)$$

and assuming

$$f = \frac{.0791}{Re^{.25}} \qquad (2.2.11)$$

then the shear force becomes

$$\tau_x = \frac{.04\rho(\omega y - V_x)^{1.74}}{(\frac{2 S}{\nu})^{.25}} \qquad (2.2.12)$$

It has been shown[5] that for steady state operation

$$\tau_x = \frac{\partial P}{\partial x} S \qquad (2.2.13)$$

and

$$V_x = \omega(y - y_{avg}) \tag{2.2.14}$$

where

$$y_{avg} = \frac{y_s + (R^2 - x^2)^{.5}}{2} \tag{2.2.14}$$

Using Equations (2.2.13), (2.2.14) and $P = \rho g h_d$, then Equation (2.2.12) becomes:

$$\frac{\partial h_d}{\partial x} = \frac{.04(\omega y_{avg})^{1.75}}{g\left(\frac{2S}{\nu}\right)^{.25} S} \tag{2.2.16}$$

or the pressure head generated by the rotating discs is:

$$h_d = 2K \int_0^{L/2} \left[\frac{y_s + \sqrt{R^2 - x^2}}{2}\right]^{1.75} dx \tag{2.2.17}$$

where

$$K = \frac{.04 \quad \omega^{1.75}}{g\left(\frac{2S}{\nu}\right)^{.25} \quad S} \quad (\omega \text{ in radians/sec})$$

Instead of integrating Equation (2.2.17) the result may be closely approximated by the summation of a finite number of sections of the submerged disc, as shown in Figure 2.2-3, where now:

$$h_d = 2K \sum_{i=0}^{n} \left(\frac{y_s + \sqrt{R^2 - X_i^2}}{2} \right)^{1.75} \Delta x \qquad (2.2.18)$$

$$X_i = \frac{L}{4n} + \frac{L}{2n}(i) \qquad i = 0,1,2\ldots n$$

$$\Delta X = \frac{L}{2(i + 1)}$$

Using Equation (2.2.18) and Table 2.2-1 the pressure head due to the rotating discs (h_d) is solved for. Tables 2.2-4 and 2.2-5 list the results for the one foot diameter and the five foot diameter discs, respectively.

Figures 2.2-2 and 2.2-3 show that the horizontal shear force model underpredicts the disc pressure head, but that it does follow the same trend as the experimental data. The friction factor assumed in Equ. (2.2.11) is a curve fit, which is only accurate for Reynolds number in the lower turbulent regime, and it would tend to be smaller than it should be, the higher the rotational speed of the discs. That would make the mathematical model stray away from paralleling the experimental values. This is illustrated in Figures 2.2-1 and 2.2-2. With this model, however, there appears to be little danger of overpredicting how much assistance the pressure head due to the rotating discs will be in curbing possible flow irregularities. Therefore this model will be used in future disc pressure head predictions.

TABLE 2.2-4

SHEARING MODEL PREDICTIONS OF PRESSURE HEADS

ACROSS ROTATING ONE FOOT DIAMETER DISCS

ω(rpm)	h_d	ΔP_s	$h_d + \Delta P_s$ (in)
5	.003	.001	.004
10	.010	.003	.013
15	.020	.006	.026
20	.033	.011	.044
30	.068	.025	.043
40	.113	.044	.157
50	.166	.070	.236

TABLE 2.2-5

SHEARING MODEL PREDICTIONS OF PRESSURE HEADS

ACROSS ROTATING FIVE FOOT DIAMETER DISCS

ω(rpm)	h_d (in)
1	.003
2	.010
3	.021
4	.034
6	.669
8	.116
10	.170

c) A Possible Prototype Disc Matrix

It was originally thought that the final periodic cooling tower would consist of five foot diameter discs, because that was the maximum size that could be made from stock sheetmetal. By going to a segmented design, Figure 2.2-4, the disc areas that were ineffective for heat transfer are removed and there is not an excessive amount of waste produced when stamping out a series of similar segments. This reduces production costs. Segmented discs also enable the discs to be made with a larger heat transfer surface per unit weight. In addition the larger discs make it possible to stack fewer modules, thus cutting down on the size of the pumps needed to pump the circulating water to the modules. It is desired to pre-fabricate these modules at the factory and ship them as complete units. This limits the module size to that which can be shipped on a rail car.

It has been estimated that the disc diameter of such a module would be 8.5 feet. Figure 2.2-5 shows the predicted pressure heads due to rotating discs of this size with a disc spacing of .5 inch.

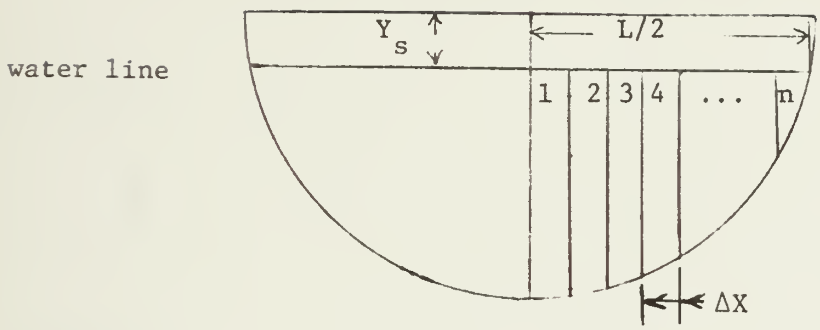


Figure 2.2-3 Illustration of the Submerged Portion of the Disc Divided into a Finite Number of Segments

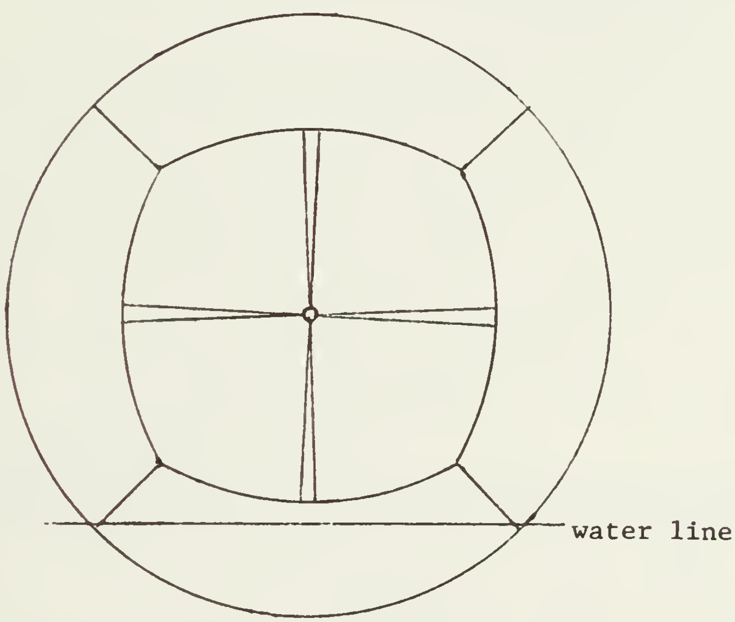


Figure 2.2-4 Segmented Disc Design with Hollow Disc Centers

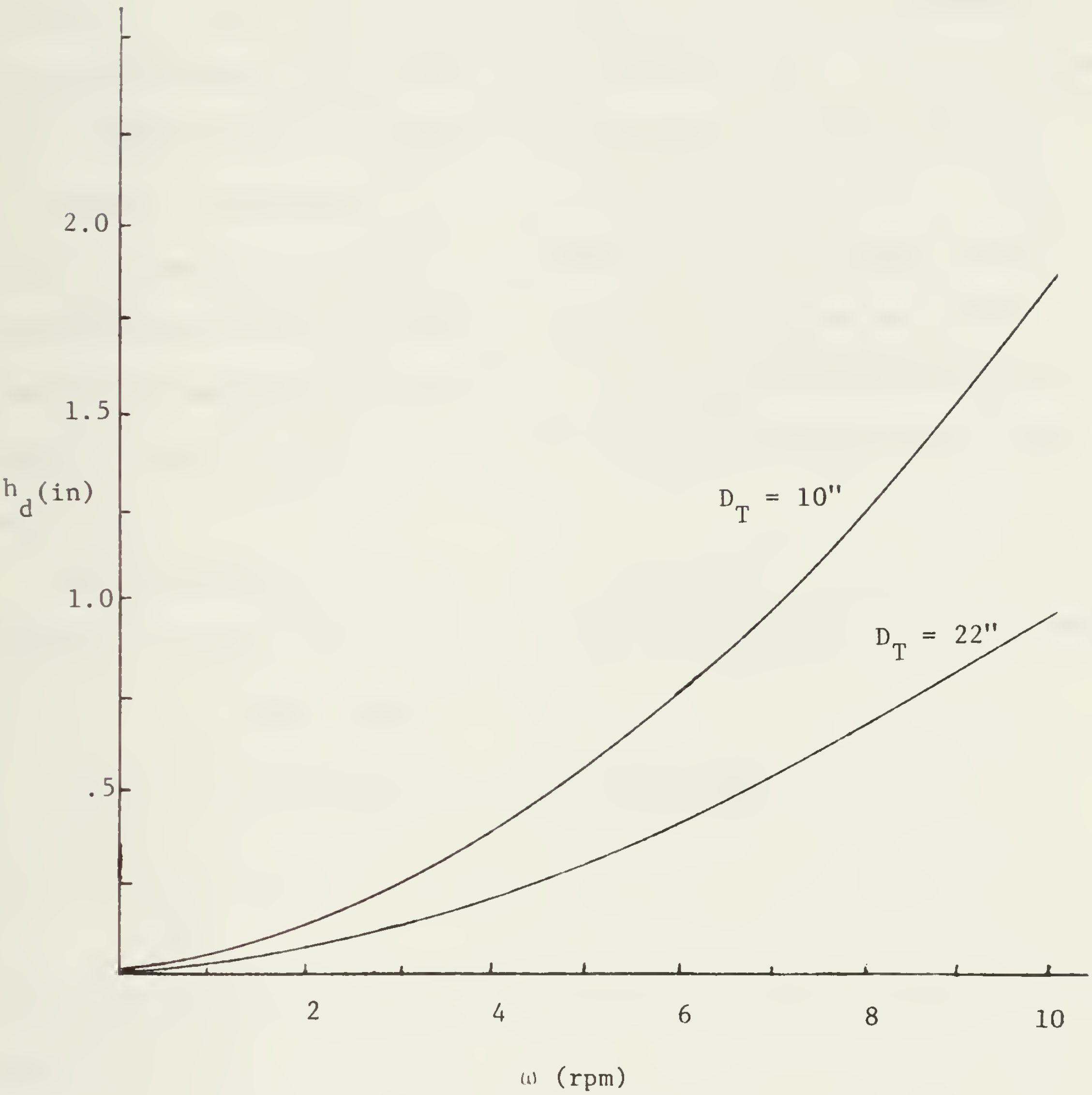


Figure 2.2-5 Predicted Pressure Heads Created by 8.5 Foot Diameter Rotating Discs for Two Different Basin Depths

2.3 Uncertainty Analysis

There was some uncertainty in the measurement of the pressure drop across the discs using the inclined manometer. Care was taken to purge the manometer and taps of trapped air prior to use. Also the manometer was cleaned to eliminate deposits from contaminants in the water from the mockup. At low disc rotational speeds the fluid in the manometer was steady and the pressure drop could be easily read. At high rotational speeds the fluid inthe manometer fluctuated and an average value had to be taken for the pressure drop. During the experiments the rotational speed was gradually increased and pressure readings taken and then the rotational speed was decreased and a new set of readings were taken. That was to demonstrate repeatability of the measured pressure drop at different rotational speeds and hence increase confidence in the data being taken.

The experimentally measured head created by the rotating five foot diameter discs was found to be, at 4 rpm (an approximate operating speed).

$$h_d = .068 \pm .005 \text{ in}$$

The error in measurement can be expressed as[6]

$$\frac{\Delta h_d}{h_d} = .07$$

So the pressure head is known to within about 7% experimentally, in the region of 4 rpm.

A rotational speed of 20 rpm for the one foot diameter discs yields a tip speed the same as that for the five foot discs at 4 rpm. The disc spacers created an additional error in pressure drop measurement and for 20 rpm:

$$h_d = .065 \pm .01 \text{ in}$$

So the pressure head is known to within about 15% experimentally, in the region of 20 rpm for the smaller discs.

CHAPTER 3

INLET AND OUTLET TROUGH SIZING

Signficant reductions in trough widths will assist in reducing the weight of the periodic cooling tower modules. Width reductions must be made with care, so that there will be an even distribution of flow to all the disc spacings. This can be accomplished by the use of orifices between the inlet trough and the basin. The orifices make it possible to increase the length of the cooling tower modules by reducing the area available for lateral discharge. The size and number of orifices required depends on several factors, which are identified in this chapter.

3.1 Diffuser Analysis

The flow distribution from the inlet trough is essentially a diffuser problem (where a diffuser is defined as a duct which distributes the water flowing through it, evenly along its length). It is necessary to distribute the water entering the basin uniformly through the disc spacings. To do this the pressure drop through the discs plus orifice, i.e. from the inlet to the outlet trough must be large enough to make the effects of the frictional head losses along the troughs insignificant. The curves in Figure 1.2-3 must be separated so that h_{min} is approximately equal to h_{max}. The rotating discs, as has been illustrated, can create some of the neededflow resistance. That can be supplemented by orifices on the disc shroud shown in Figure I-1. There the shroud is a means of separating the basin from the inlet and outlet troughs. The orifices must be well oriented with respect to the disc spacing to ensure that there are no areas in theinlet trough that are starved of circulating water flow.

Discharge through an orifice from a reservoir can be described by[7]:

$$Q = C_d A_o \sqrt{2gH_o} \tag{3.1.1}$$

where

Q = flow rate (ft^3/sec)

C_d = discharge coefficient

A_o = orifice area (ft^2)

H_o = head above the center of the orifice (ft)

The discharge coefficient is not a constant for orifices along a laterally discharging diffuser, but a function of the static head above the center and the velocity head. The velocity head is defined as[7]:

$$h_v = \frac{Q^2}{2gA_R^2} \tag{3.1.2}$$

where

A_R = the cross-sectional area of the trough

Equation (3.1.2) with the velocity head in inches of water may be written (for T = 135°F) as:

$$h_v = 9.27 \times 10^{-7} \left[\frac{N\ L_T}{A_R}\right]^2 \tag{3.1.3}$$

where

L_T = the trough length to the point of interest (ft)

N = the number of gallons of flow per minute per foot of trough.

Rawn[8] has developed curves that relate the discharge coefficient to the ratio of h_v to H_o. For a sharp-edged orifice the equation

$$C_d = .61\ (1 - \frac{h_v}{H_o}) \quad 0 \leq \frac{h_v}{H_o} \leq .5 \qquad (3.1.4)$$

fits the curve within the limits specified.

Examination of Equations (3.1.1) and (3.1.4) reveals that as the velocity head increases then the discharge through an orifice decreases. Therefore, to maintain a constant flow rate along each portion of the basin, it may be necessary to increase the size of the orifices the nearer they are to the beginning of the inlet trough, when the inlet trough is narrow.

However, if the width of the inlet trough is large enought that the discharge coefficient is a constant then it may be necessary to decrease the size of the orifices the nearer they are to the beginning of the trough. This is because of the added contribution of the frictional head near the inlet.

3.2 Frictional Head Losses Through a Continuous Laterally Discharging Diffuser

Equation (2.2.1) can be used to determine the frictional head losses along the trough length. Now, assuming the hydraulic diameter concept is valid for the trough geometry, then the local frictional head is:

$$\Delta P = 4f \left(\frac{LT}{D_e}\right)\left(\frac{\rho V^2}{2}\right) \tag{3.2.1}$$

where

$$V = \frac{\dot{W}}{\rho A_R} \tag{3.2.2}$$

$$D_e = \frac{4A_R}{P} \tag{3.2.3}$$

The length along the trough to the point of interest is L_T (ft). the cross-sectional area of the trough is A_r (ft^2) and its wetted perimeter is P (ft). The mass flow rate through the trough is $\dot{W}$ and it is in units of lbm/hr. Using Equations (3.2.2) and (3.2.3) Equation (3.2.1) becomes (P in inches of water):

$$\Delta P = 2.30 \times 10^{-10} \frac{f\ L_T\ \dot{W}^2\ P}{\rho\ A_R^3} \tag{3.2.4}$$

Equation (3.2.4) can be solved for the total frictional head loss over the entire trough. First an equation that will generate the proper friction factor must be found. For Reynolds numbers in the range

$$2.30 \times 10^3 \leq Re \leq 3.44 \times 10^5$$

and $\frac{e}{D_e}$ ratios in the range of:

$$.00022 \leq \frac{e}{D_e} \leq .0015$$

then the equation for friction factor

$$f = \frac{.0791}{Re^{.25}} + 3.762 \times 10^{-4} \left(\frac{e}{De}\right) (Re)^{.3} \qquad (3.2.5)$$

closely duplicates Moody's graph of friction factor for different Reynolds numbers[4].

Letting $\dot{W} = L_T \dot{W}_f$, where $\dot{W}_f$ is the mass of water that is removed laterally per foot of inlet trough, then Equations (3.2.4) can be integrated over the entire trough length (ℓ) to find the head losses due to friction.

$$h_f = \frac{1.286 \times 10^{-11} (\nu^{.25} P^{1.25} \dot{W}_f^{1.75})}{\rho^{.75} A_R^3} \int_0^{\ell} L_T^{1.75} \, dL_T$$

$$+ \frac{8.073 \times 10^{-14} (e^{.35} P^{1.03} \dot{W}_f^{2.3})}{A_R^{3.35} \rho^{1.3} \nu^{.3}} \int_0^{\ell} L_T^{2.3} \, dL_T \qquad (3.2.6)$$

At a trough water temperature of 135°F and with $\dot{W}_f = (4.92 \times 10^2)N$ lbm/hr-ft Equation (3.2.6) becomes:

$$h_f = 4.23 \times 10^{-9} \left[\frac{P^{1.23} \ell^{2.75} N^{1.75}}{A_R^3}\right]$$

$$+ 5.67 \times 10^{-10} \left[\frac{e^{.35} P^{1.03} \ell^{3.3} N^{2.3}}{A_R^{3.35}}\right] \qquad (3.2.7)$$

To find the head loss up to a specific point along the inlet trough, Equation (3.2.6) is integrated to that point. If, instead, the head loss to that same point needs to be known on the outlet trough then the result of integrating Equation (3.2.6) must be subtracted from the h_f calculated for the entire trough length. Performing those steps will yield a graph similar to that shown in Figure 1.2-3.

The proposed periodic cooling tower with 8.5 foot diameter discs, a trough water depth of 22 inches and a flow rate of 25 GPM/ft, will have head losses along the trough as shown in Figure 3.2-1.

The trough width (W_T) in Figure 3.2-1 is the distance from the edge of the disc to the trough wall at the waterline. Even with this distance as small as .5 feet the cross-sectional area of the trough is large enough to make the frictional head losses small for all but very long troughs.

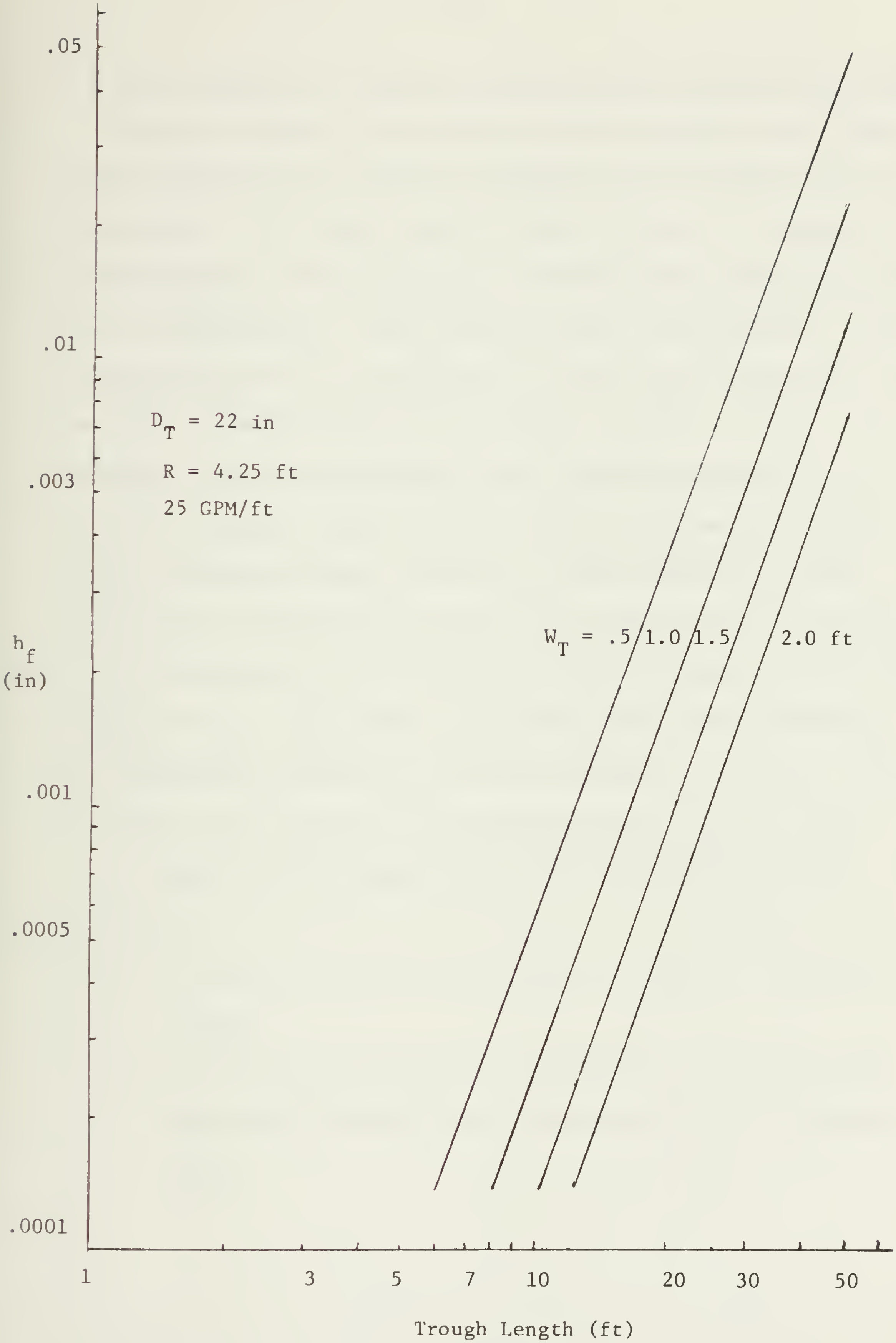


Figure 3.2-1 Frictional Head Losses Along a Latterally Discharging Inlet Trough

3.3 Orifice Sizing for Equal Distribution of Flow Into the Disc Spacings

Just the rotation of the discs in the basin might create a large enough pressure head to make the magnitude of the frictional head loss insignificant. That would make the curves in Figure 1.2-3 appear to be approximately straight lines and, it would seem, eliminate the chance of flow irregularities. But is has been found from experience that the cross-sectional area of a diffuser inlet should not be exceeded by the sum of the aggregate areas of the discharge spaces[8]. In fact, it is necessary to have cross-sectional area considerbly larger than the sum of the aggregate areas to ensure that the orifices flowfull. A survey of diffuser designs[7] reveals that most utilize only about 65-75% of the cross-sectional area for the total allowed orifice area, with an occasional use of as low as 20% of the area and as high as 80%. An inlet trough of any useable length will have a lateral discharge area considerably larger than its cross-sectional area, unless orifices are used in the disc shroud separating the inlet trough and basin.

A number of things must be kept in mind when making orifice selections.

1. The orifice must be large enough to minimize the chances of fouling.
2. A good orifice pattern must be selected to ensure that the disc matrix receives a well distributed flow of circulating water.

3. When comparing specific designs, the volume of water gained by increasing the head due to smaller orifices should be less than that removed by reducing the trough width. Also the supporting structure gained should be less than that removed by reducing the trough width. From Figure 3.3-1:

$$A_{R2} < A_{T1}$$

$D_{T2} - D_{T1} < W_{T1} - W_{T2}$ (assuming the structural weight is the same in all directions).

4. Assuming a plant efficiency of 40 percent, a temperature reduction of 20°F for the circulating water as it passes through the periodic cooling tower and a 1000 MegaWatt plant, it can be shown that the increase in pumping power when the inlet trough is reduced is negligible when compared to the plant output. Therefore the orifice sizes won't make a noticeable impact on circulating pump requirements.

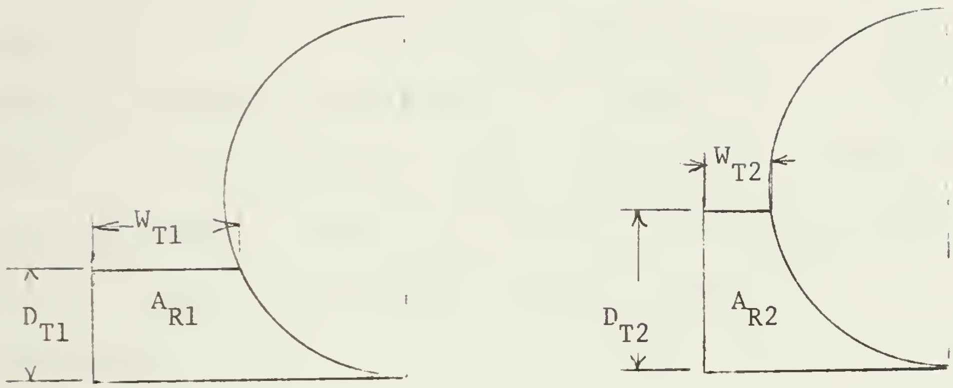


Figure 3.3-1 The Variation of Trough Size as the Trough Width Varies

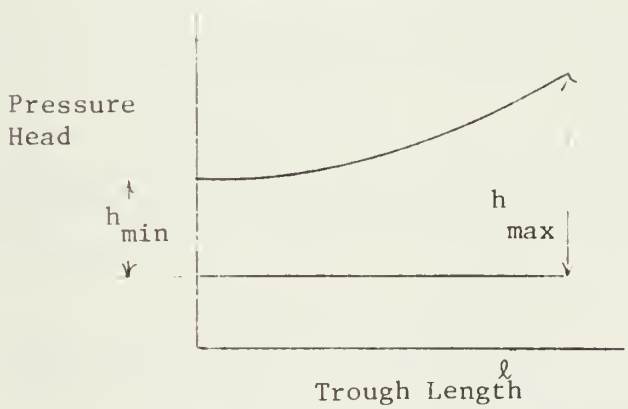


Figure 3.4-1 Illustration of the Pressure Curves for a Diffuser Discharging into a Reservoir

3.4 Extreme Cases in Disc Matrix Orientation

There are two extreme cases for the orientation of the disc matrix with respect to the orifices. The first case is where the clearance between the discs and the troughs are so large that the basin is similar to a large reservoir with a constant water level throughout. Figure 3.4-1 shows the relationship between the head along the inlet trough with respect to the basin. The minimum head is at the far end of the inlet trough and is composed of

$$h_{min} = h_o + h_d \qquad (3.4.1)$$

The maximum head is at the beginning of the inlet trough and is

$$h_{max} = h_f + h_o + h_d \qquad (3.4.2)$$

The second extreme case is where the clearance between the shroud and the discs is very small. This is represented in Figure 3.4-1. Here the maximum head (with orifices between the basin and both the inlet and outlet troughs) is;

$$h_{max} = 2h_o + h_d + h_f \qquad (3.4.3)$$

The minimum head is located half-way down the module, due symmetry, and is:

$$h_{min} = h_{max} - (h_f - h_{f/2}) + h_{f/2} \qquad (3.4.4)$$

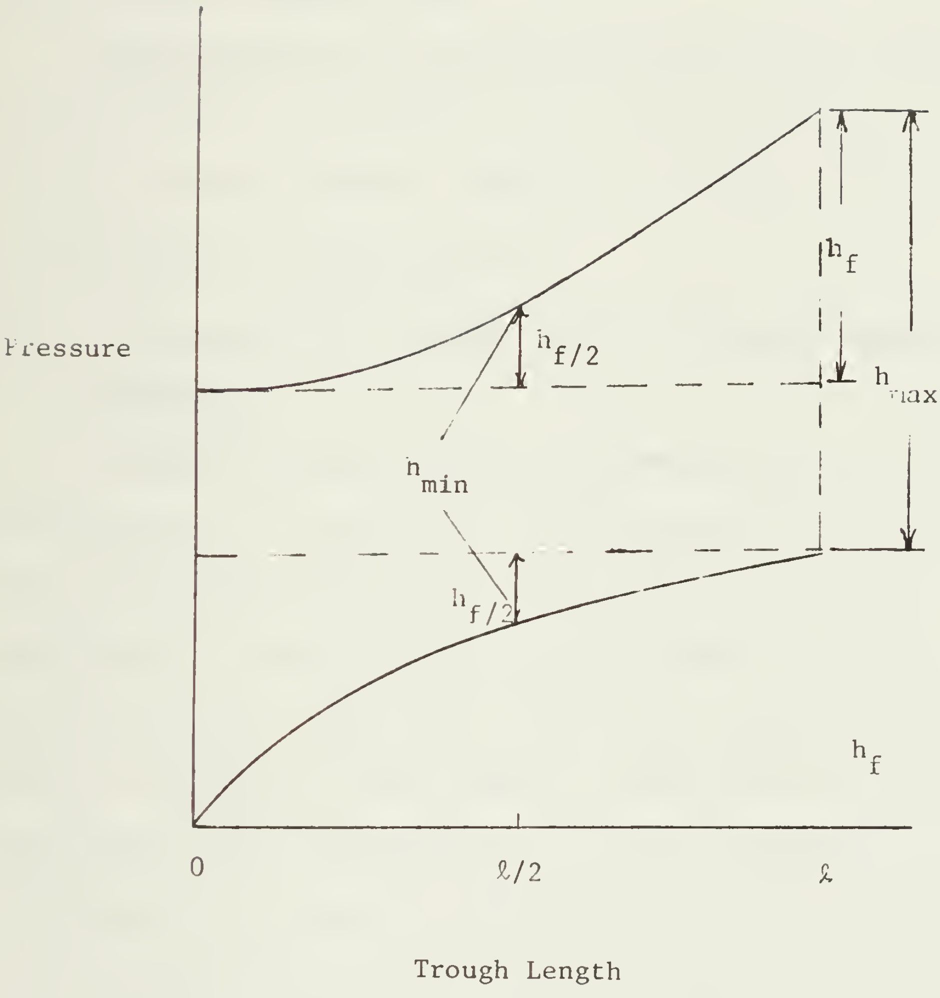


Figure 3.4-2 Pressure as a Function of Trough Length for the Case Where the Clearance Between the Shroud and Discs is Very Small

where $h_{f/2}$ is the frictional head loss half way down the trough. Combining Equations (3.4.3) and (3.4.4) yields

$$h_{min} = 2h_o + h_d + 2h_{f/2} \tag{3.4.5}$$

for the close clearance case.

When designing the flow distribution system of a periodic cooling tower it is desirable to keep the ratio of h_{max} to h_{min} as near to one as possible. That will ensure that no flow irregularities are likely to occur.

3.5 A Flow Distribution Design Based on A Proposed Segmented Disc Matrix

A proposed periodic cooling tower module consists of 8.5 foot diameter segmented discs spaced .5 inches apart in a trough 15 feet long. The depth of the basin is 22 inches and the width at the waterline is 10 inches, so the sum of the widths of the inlet trough, basin and outlet trough is 8.5 feet. The correct disc rotation rate is a function of such things as: water level; circulating water flow rate; the possible chance of oil emulsification from churning; and air velocity. The optimum rotational speed must be found experimentally, but will probably be in the neighborhood of 4 rpm for discs of this size.

In this design it is assumed that a six inch trough width at the waterline is more acceptable than the 10 inch proposed width. It is desired to determine what size orifices would be

required to ensure that each portion of the basin receives the same flow rate. Condenser tube fouling experiments are being conducted on .75 inch inside diameter tubes and that can be used as a minimum acceptable diameter for an orifice.

At the far end of the trough both the velocity head and head losses due to friction will be negligible. Equation (3.1.1) can be rewritten so that the head caused by the restriction of flow through the orifice is:

$$h_o = 3.11 \times 10^{-2}\left[\frac{N}{n\, C_d d_o^2}\right]^2 \qquad (3.5.1)$$

where h_o is in inches as is d_o (the orifice diameter) and the water temperature is assumed to be approximately 135°F. The number of orifices per foot of trough length is n. Table 3.5-1 shows what h_o will be for a number of different orifices sizes at a flow rate of 25 GPM/ft. A good flow pattern can be maintained to all the disc spaces along the trough if the product of the number of orifices per foot and their diameters is one foot. To do that the orifices must be offset enough to avoid any interaction with each other. The shroud is large, so the orifices can be spaced to avoid that problem.

The cross-sectional area of the inlet trough is found, using Equation (C-1), to be 2.82 ft^2 for the given trough dimensions. The sum of the individual areas are also shown in Table 3.5.1. Requiring that the sum of the orifice areas be seventy percent of the

TABLE 3.5-1

POSSIBLE ORIFICE SIZES WHICH MAY BE APPLICABLE TO THE PERIODIC COOLING TOWER

d_o	n	h_o	Total Orifice Area (ft^2)
.75	16	.645	.74
1.0	12	.363	.98
1.5	8	.161	1.47
2.0	6	.091	1.96
3.0	4	.040	2.94

TABLE 3.5-2

TABULATION LEADING TO THE FLOW RATE DELIVERED TO THE DISCS ALONG VARIOUS LENGTHS OF TROUGH

ℓ	h_v	h_f	$H_o=h_f+h_o$	$\frac{h_v}{H_o}$	C_d	N GPM/ft
.5	1.8×10^{-5}	1.40×10^{-7}	$\simeq .645$	$\simeq 0$	.61	25.0
7.5	4.10×10^{-3}	2.54×10^{-4}	.645	.006	.61	25.0
14.5	1.53×10^{-2}	1.60×10^{-3}	.647	.024	.60	24.6
29.5	6.34×10^{-2}	1.18×10^{-2}	.657	.097	.55	22.7
49.5	1.79×10^{-1}	5.12×10^{-2}	.696	.256	.45	19.2

cross-sectional area limits the orifice size to a maximum of 2 inches in diameter for this trough.

Figure Figure 2.2-5 the pressure head created by the disc rotating at 4 rpm is .196 in. The velocity head is then found by using Equation (3.1.3) and the head losses due to friction by either Equation (3.2.7) or Figure 3.2-1. The results are shown in Table 3.5-2 for various trough lengths besides 15 feet.

Examination of the two extreme cases in disc orientation reveals that the ratio of Equation (3.4.2) to Equation (3.4.1) is larger than the ratio of Eqaution (3.4.3) to (3.4.5). The true disc matrix position in the trough is actually somewhere between the two extremes. The most conservative case is when there is a large clearance between the matrix and the shroud and is the one considered in this example.

Table 3.5-2 shows the flow rates through .75 in diameter orifices along various trough lengths. The discharge coefficient was found using Equation (3.1.4). The flow rate through the oricies per foot of trough length is now

$$N = 5.67\ n\ C_d\ d_o^{\ 2}\ H_Q^{\ .5} \qquad (3.5.2)$$

The .75 inch orifices do an adequate job of distributing the low to the basin for the desired length of 15 feet. Byond that the flow distribution becomes progressively worse, as evidenced by the decreasing values of N. Larger orifices only aggravate the problem. While the .75 inch orifices do distribute the flow satisfactorily they raise the trough depth from 22 inches to 22.647 inches. That changes the trough cross-sectional area slightly but has a negligible

effect on the lateral flow rate per foot of trough. However it does decrease the value of h_f from 1.60×10^{-3} in to 1.47×10^{-3} in, but that only changes the ratio of h_{max} to h_{min} from 1.0019 to 1.0017, an exact solution could be obtained by iteration. In either case that is significantly closer to one that the 1.1 that is considered the minimum necessary to ensure even flow. Table 3.5-3 shows a summary of the periodic cooling tower module's dimensions and the predicted flow parameters.

The design reduced the cross-sectional area of the inlet trough, reducing the waterweight and also reducing the structural weight from what was initially proposed. The selected orifices also ensure that there will be no flow irregularities in the design.

TABLE 3.5-3

CHARACHTERISTICS OF THE FLOW DISTRIBUTION SYSTEM

OF A PROPOSED PERIODIC COOLING TOWER MODULE

Disc Diameter	8.5 ft
Disc Spacing	.5 in
Trough Length	15.0 ft
Trough Depth	22.65 in
Trough Width	6 in
Flow Rate	25 GPM/ft
Number Orifices/ft	16
Diameter Orifices	.75 in
Water Temperature	135°F
h_{max}/h_{min}	1.002

CHAPTER 4

THE USE OF WEIRS FOR WATER LEVEL CONTROL

It may be necessary to allow the flow rate of water along the trough to fluctuate as the demands on the cooling system vary. It would be desirable to maintain the optimum water level in the basin as nearly as possible. That would ensure that heat transfer is maximized and also keep the oil layer from being washed away into the rest of the system. The use of weirs for water level control is an attractive possibility. There are no moving parts, they should be inexpensive to construct, and require little maintenance.

4.1 The Superiority of the Rectangular Weir in Water Level Control

The rectangular weir is only one passive means of water level control. It is possible to use orifices, sluice gates, and V-notch weirs. For orifices and sluice gates the flow rate is proportional to square root of the head, while for a rectangular weir the flow rate is proportional to the three-halves root of the head. This means that for a given increase in flow rate the rectangular weir will pass the same amount of water with a smaller change in head than either the orifice of sluice gate. A V-notch weir is shaped as its name implies and, because of its changing cross-section, its head is more sensitive to fluctuations of flow rates than is the rectangular shaped weir. Therefore the rectangular shaped weir appears to be best suited for application in the periodic cooling tower.

4.2 Description of Suppressed and Unsuppressed Weirs

The bottom edge of the rectangular weir is called the crest (see Figure 4.2-1), while the sides are known as weir ends. A sharp crested weir has a crest edge of between .03 and .08 inches thick[9]. The water flow through the weir is known as the nappe. There is a small drop in water level that starts up stream of the weir that is caused by the acceleration of the water as it comes nearer the crest. That drop is known as the drawdown. The water level of the downstream side of the channel must be low enough so that the nappe will be fully ventilated with air. If it isn't then the effect of low pressure under the nappe will be to increase the flow rate and the weir formulas won't be of use.

A large approach pool allows the water to approach the crest laterally as well as perpendicularly. The lateral approaching water also accelerates as it nears the weir ends, therefore it can't make the 90° turn immediately after it reaches the end, and a contraction in the nappe results. This effect can occur due to water approaching from the bottom of the pool also. An unsuppressed weir has a contracted nappe. A suppressed weir has its sides or the pool bottom very close to the crest so that there is no chance for the water to develop an appreciable lateral momentum and therefore there isn't any nappe contraction. A suppressed weir often needs to be artificially ventilated by a pipe to the atmosphere because the channel downstream of the weir is the same width as the nappe.

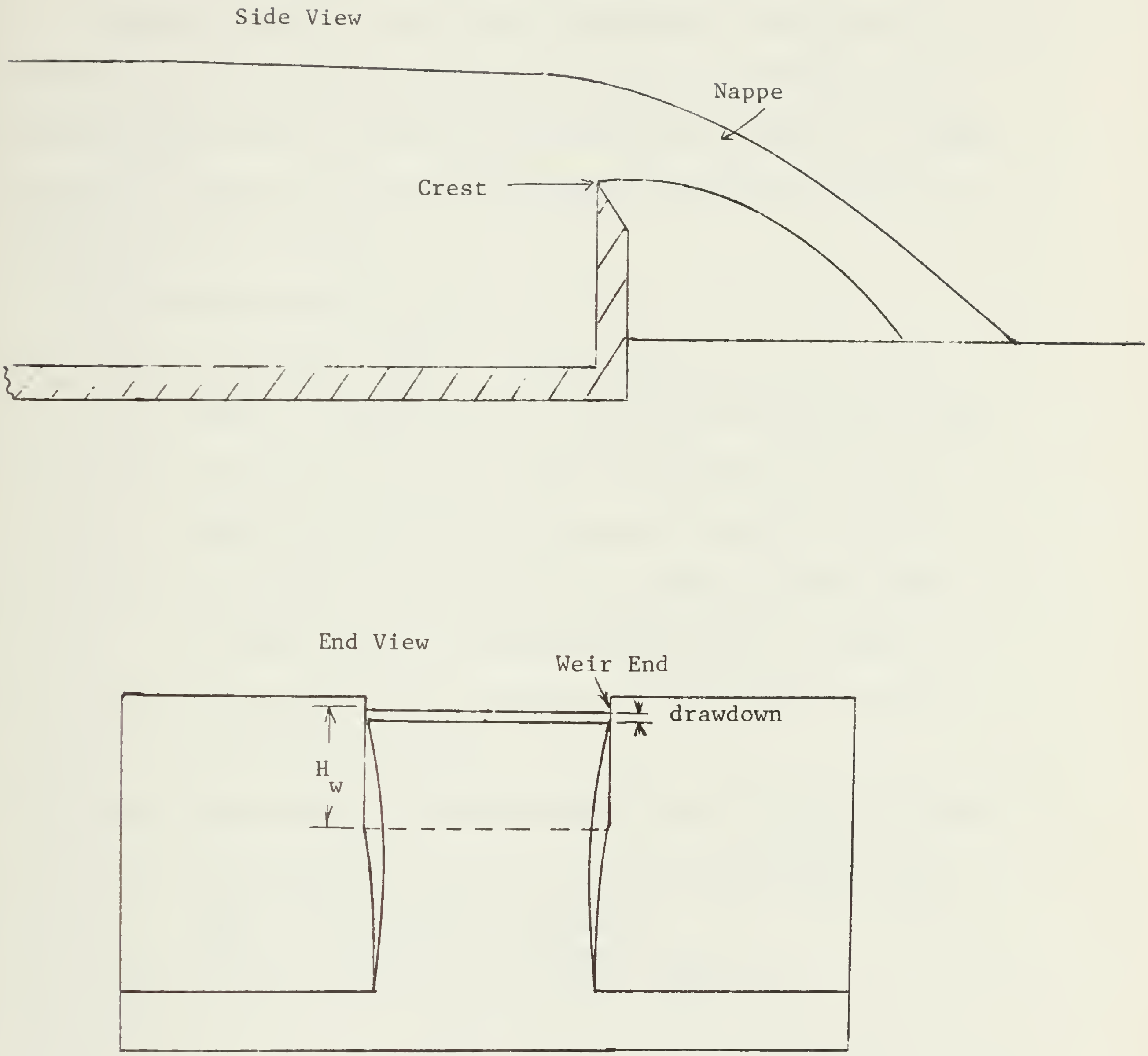


Figure 4.2-1 Unsuppressed Rectangular Weir

4.3 Restrictions in the Use of Weir Formulas

There are a number of dimensional restrictions which must be adhered to if the weir formulas are to be used to accurately predict the heads that will be developed for different flow rates[9]. The head is shown in Figure 4.2-1 and is the distance from the weir crest to the water level in the pool.

For a fully contracted weir:

1. The distance from the channel sides to the weir ends should be greater than 2.5 H_w and not less than one foot.
2. The distance from the crest to the bottom of the pool should be greater than 2 H_w and not less than one foot.
3. The cross-sectional area of the approach channel must be eight times the cross-sectional area of the nappe at the crest, for a distance of 15-20 times the depth of the nappe, upsteam.

For either a suppressed or unsuppressed weir that is fully ventilated

4. The minimum head must be a .2 feet (this prevents the nappe from clinging to the crest).
5. The crest length must be greater than 3 H_w.

4.4 Possible Adaption of the Weir for Use in the Periodic Cooling Tower

The geometry of the trough developed in Chapter 3 makes it difficult to adhere to all the restrictions outlined in Section 4.3. For purposes of demonstrating the effects of flow rate change on the head (and therefore trough water level), the Francis[9] Formulas are satisfactory.

For un-suppressed rectangular weirs:

$$Q = 3.33\ [(H_w+h_v)^{3/2} - h_v^{3/2}](L_c-0.2H) \qquad (4.4.1)$$

where

Q = flow rate (ft^3/sec)

H_w = head on the weir in feet

h_v = head due to the velocity of approach (ft)

L_c = crest length (ft)

For suppressed rectangular weirs:

$$Q = 3.33\ L_c[(H_w+h_v)^{3/2} - h_v^{3/2}] \qquad (4.4.2)$$

The 15 foot long troughs discussed in Chapter 3 had a cross-sectional area of 2.82 ft^2 and 375 GPM flowed through that area, so the velocity head, h_v, is 1.37×10^{-3} ft. The minimum allowable weir head is .2 feet, thus the velocity head can be neglected. Then Equation (4.4.1) becomes:

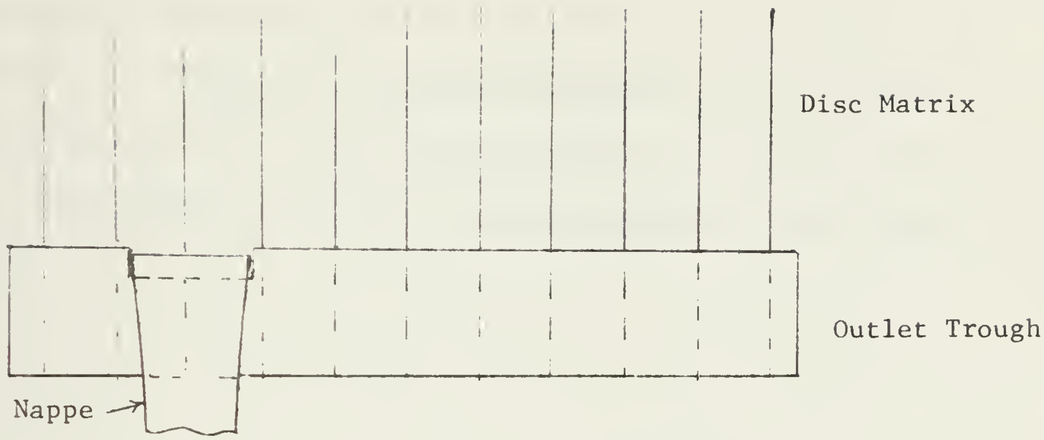
$$Q = 3.33\ H_w^{3/2}(L_c-0.2\ H_w) \qquad (4.4.3)$$

and Equation (4.4.2) becomes:

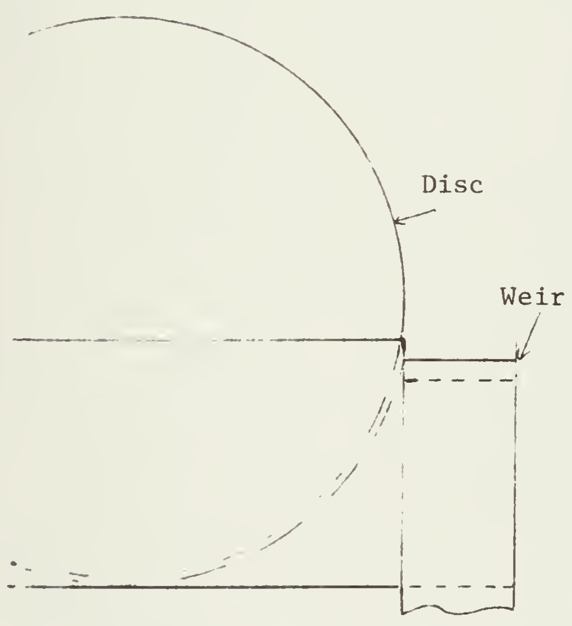
$$Q = 3.33\, L_c H_w^{3/2} \qquad (4.4.4)$$

The trough width selected in Chapter 3 was only half a foot, so an unsuppressed weir cannot be used at the end as was shown in Figure I.-1. Assuming that restriction number 3 can be satisfied for a weir position as shown in Figure 4.3-1, with a head of .23 feet, then Equation (4.4.3) can be used to find a crest length of 2.29 ft. Then as the flow rate fluctuates from 30 GPM/ft to 20 GPM/ft the head changes from .25 feet to .20 feet. Thus for a sizeable change in flow rate the water level in the trough stays within ± .36 inches. This weir would have to have one end at least a foot from the end of the outlet trough. However a weir in this position could effect the flow distribution that has been established by the rotating discs and the trough orifices by drawing off water primarily from the last few feet of the basin.

The alternative is to go to a suppressed weir with a crest length the same as the trough width: .5 feet. Then using Equation (4.4.4) as the flow rate ranges from 30 GPM/ft to 20 GPM/ft the head fluctuates from .71 ft to .54 feet. However these values exceed the restriction that the crest length be greater than three times the head. For a .5 foot crest, then the head should only be .167 feet, but this is less than the minimum .2 feet needed to insure that the nappe springs free of the weir. If it is assumed that allowing the crest length to be less than three times the head only causes errors in the specific values of the heads measured, and not in how much they change for a



(a) Unsuppressed weir
(side veiw of trough)



(b) Suppressed weir (end view of trough)

Figure 4.3-1 Possible weir positions on the Outlet Trough

fluctuation in flow rate, then the range of $\pm$ 1.02 inches is satisfactory. Clearly this assumption would have to be verified experimentally, but a .5 foot suppressed weir at the end of the outlet trough does appear to be a suitable method of water level control.

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

A well distributed flow of circulating water through the disc matrix is necessary to help ensure maximum heat transfer is obtained. The desired distribution can be obtained if the pressure difference across the discs is large compared to the pressure difference through the inlet trought.

Two simple methods were developed to predict the pressure head developed by the rotating disc matrix. Comparing their predictions to experimentally derived pressure drops, it was found that the horizontal shear force model was the most conservative. The model was then used to estimte the head developed by a proposed 8.5 foot diameter segmented disc matrix.

The inlet and outlet troughs will most likely have a constant cross-sectional area throughout their lengths. That will aid in modular construction and simplify erection. Calculating the head losses alongthe troughs then becomes a diffuser problem. The frictional head losses were calculated for a laterally discharging diffuser with the geometry of the troughs as described. In these calculations the friction factor was not a conservatively selected constant, but instead changed as the Reynolds number decreased on progressing down the trough.

In addition to restricting the flow to basin enough to make the changes in the frictional head along the inlet trough neglibible, it is necessary to keep the total area through which the water is later-

ally discharging, smaller than the cross-sectional area of the trough. Both of these criteria can be met with the use of orifices in the shroud. A periodic cooling tower module has been proposed which will be 15 feet long and contain 8.5 foot diameter discs, .5 inches apart. It was demonstrated that orifices placed in the shroud between the inlet trough and basin would provide an even flow distribution of circulating water to the disc spacings, while allowing the trough widths to be reduced to half a foot. The modest increase in pressure head will have a negligible effect on pumping power requirements.

The weir has been proposed as a simple, easily maintained, water level control device. The geometry involved in a periodic cooling tower module violates some of the requirements for successful application of weir formulas to prediction of flow rates. Assuming that the formulas continued to predict the correct variation of flow rate with changes in water levels, it was demonstrated that a supressed weir at the end of the outlet trough would still be suitable for water level control. This assumption ought to be verified experimentally with a mockup of the same geometry for the approach pond as that in the outlet trough.

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APPENDIX A

EQUIVALENT FLAT PLATE DEPTH

A suitable hydraulic diameter has in to be established in order to use Equation (3.2.1). This was done by reducing the submerged portion of the disc to an equivalent area flat plate, with the same waterline length. The depth of this flat plate is Y_e, which is used to calculate D_e.

From the geometry of Figure A-1:

$$\frac{L}{2} = \sqrt{R^2 - Y_s^2}$$

$$\gamma + \theta + \phi = 90°$$

$$Y_e = R\cos\theta - Y_s$$

Area A must be equal to Area B

$$\text{Area A} = (Y_e+Y_s)(\tfrac{L}{2}) - \frac{\gamma}{360}\pi R^2 - \frac{Y_s L}{4} - \frac{1}{2}(Y_e+Y_s)(R\sin\theta)$$

$$\text{Area B} = \frac{\theta}{360}\pi R^2 - \frac{1}{2}(Y_e+Y_s)(R\sin\theta)$$

or

$$\frac{\theta}{360}\pi R^2 = (Y_e+Y_s)(\tfrac{L}{2}) - \frac{\gamma}{360}\pi R^2 - \frac{1}{2}(Y_s)(\tfrac{L}{2})$$

$$\frac{\pi R^2}{360}(90-\phi) = \frac{LR}{2}\cos\theta - \frac{Y_s L}{4}$$

$$\cos\theta = \frac{2}{LR}\left[\frac{\pi R^2}{360}(90-\phi) + \frac{Y_s L}{4}\right]$$

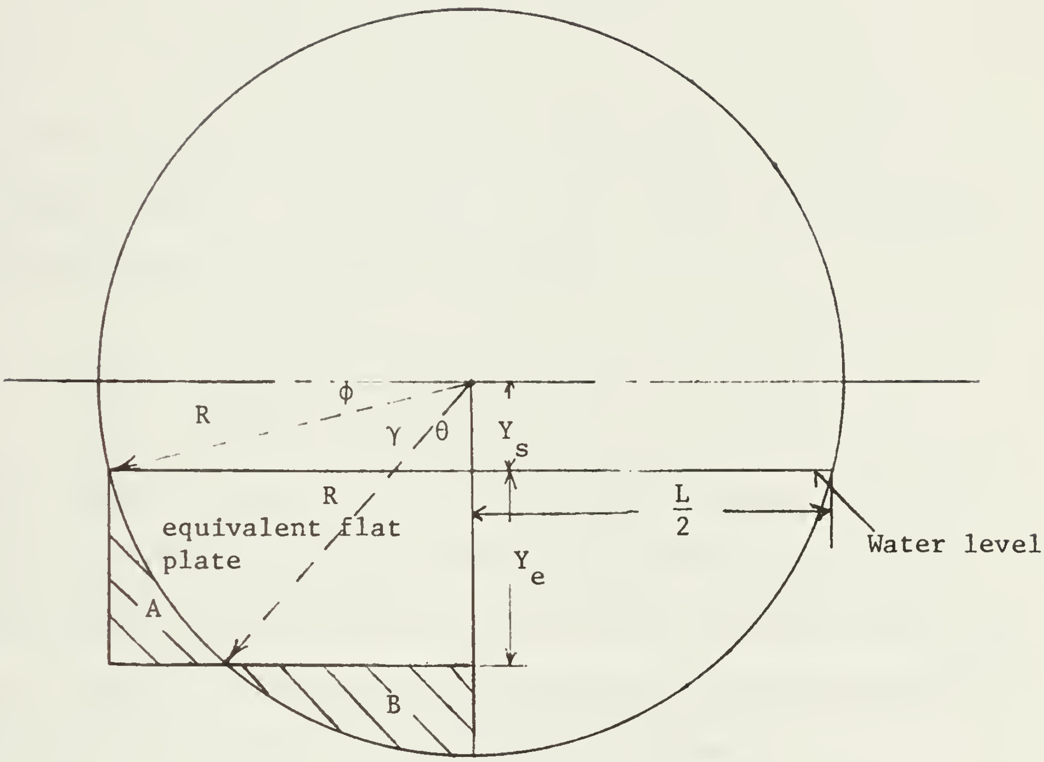

Figure A-1 Reduction of the Submerged Portion of the Disc to an Equivalent Flat Plate

but

$$\phi = \sin^{-1} \frac{Y_s}{R}$$

and so

$$Y_e = \frac{\pi R^2}{360\sqrt{R^2 - Y_s^2}} (90 - \sin^{-1} \frac{Y_s}{R}) - \frac{Y_s}{2}$$

The Y_e was calculated for an equivalent flat plate for both the 1 foot diameter and the 5 foot diameter discs. That Y_e was then used in the calculation of hydraulic diameters.

$$Y_{e1} = .26 \text{ feet (with } Y_s = .152 \text{ ft, } R = .5 \text{ ft)}$$

$$Y_{e5} = .98 \text{ feet (with } Y_s = 1.13 \text{ ft, } R = 2.5 \text{ ft)}$$

The submerged portion of the wheel could be broken into more than just one plate, the sum of which would yield the same area. This generates a pressure head across the discs somewhat higher than before. However, the crudeness of this method of approximating the head would make it hard to justify such a refinement.

A single flat plate was also selected for its adaptability to the segmented disc shown in Figure 2.2-4.

APPENDIX B

ESTIMATING THE PRESSURE DEVELOPED BY SPACER DRAG FORCES

Unfortunately the one foot diameter discs had spacers that were dragged through the water as the disc matrix rotated. This required than an additional pressure due to the spacers (ΔP_s) be added to that already calculated for the discs alone. A rough estimate of the magnitude of the pressure was made.

The spacers were .25 inches thick and .38 inches in diameter. Figure 2.1-1 shows how they were arranged in the disc matrix. The distance from the center to the spacers was:

$$R_s = 5 \text{ inches.}$$

With $Y_s = 1.8$ inches, from the geometry illustrated in Figure B-1, there is only one set of spacers submerged as the discs rotate from a to b and

$$\theta = 21°$$

The average velocity of the spacer between and b is:

$$V_{2av} = \frac{V_{ab} + 2\pi R_s \omega}{2}$$

where V_{ab} is the x-component of velocity of the spacer just as it enters or leaves the ab zone.

$$V_{ab} = 2\pi R_s \omega \cos\theta$$

The average velocity of the spacer between the waterline and a or b is:

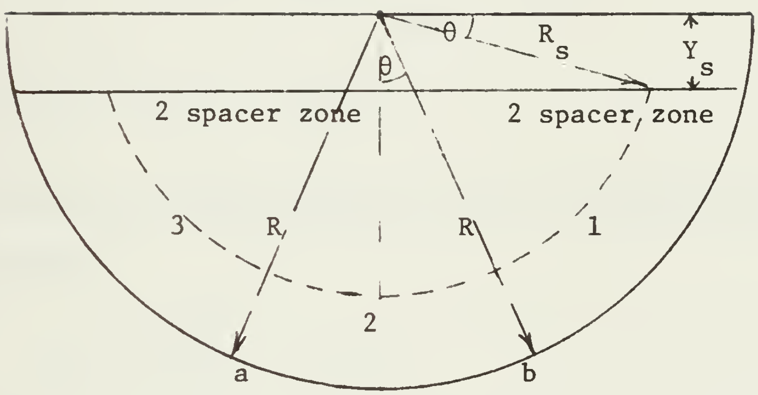

Figure B-1 Disc Spacer Geometry on the One Foot Diameter Discs

$$V_{1av} = V_{3qv} = \frac{V_{ab} + V_{x1}}{2}$$

where V_{x1} is the velocity of the spacer in the x-direction as it enters or leaves the water.

$$V_{x1} = 2\pi R_s \omega \sin\theta$$

Using the R_s and θ give

$$V_{2av} = 2.53\ \omega\text{ft}\ (\omega \text{ in rps})$$

$$V_{1av} = V_{3av} = 1.69\ \omega\text{ft}$$

With these average velocity values an average Reynolds number can be found

$$Re_2 = \frac{V_{2av} D}{\nu} = 1.46\ \omega\text{hr}$$

$$Re_1 = Re_3 = .98\ \omega\text{hr}$$

Now as the discs are rotated from 5 to 50 rpm the Reynolds number varies from 294 to 4380. The average drag coefficient (C_D) over this range of Reynolds numbers is 1.0 (it doesn't vary much)[4].

The drag force is:

$$F_D = \frac{C_D\ \rho\ V_{av}^2\ A}{2\ g} \quad \text{(A is the spacer projected area)}$$

and so for one spacer:

$$F_{D1} = F_{D3} = 1.80 \times 10^{-3}\ \omega^2 \text{lbb}$$

$$F_{D2} = 4.02 \times 10^{-3}\ \omega^2 \text{lbb}$$

Out of the 180° of rotation 47% of the time only one spacer is in the water and 53% of the time 2 spacers are in the water. So an estimate of the average drag force would be:

$$F_{Dav} = 2(.53F_{D1}) + .47(F_{D2})$$

$$= 3.80 \times 10^{-3}\ \omega^2 \text{lbb}$$

And finally the pressure generated by the spacers is:

$$\Delta P_s = \frac{F_{Dav}}{A_T} \quad \text{(psi)}$$

where A_T is the cross-sectional area of the immersed gap between the plates. Here $A_T = 1.05 \text{ in}^2$

$$\Delta P_s = 2.78 \times 10^{-5}\ \omega^2 \text{(inches of water,} \omega \text{ in rpm)} \qquad \text{(B-1)}$$

Using this equation the values listed in Tables 2.2-2 and 2.2-4 were derived for P_s at various rotational speeds.

It should be noted that in this approximation of the drag forces on the spacers, the water has assumed to be at rest, when in fact the rotating discs do induce water circulation.

APPENDIX C

THE HYDRAULIC DIAMETER OF A TROUGH

Figure B-1 shows the geometry of the end of either an inlet or an outlet trough. From that is can be seen:

Wetted Perimeter $= P = D_T + W_T + I + C$

$$C = \frac{2\pi R\theta}{360} \quad (\theta \text{ in degrees})$$

and

$$P = D_T + W_T + R\sin\theta + \frac{2\pi R\theta}{360}$$

The cross-sectional area is:

$$A_R = D_T(W_T + I) - [\frac{\pi R^2\theta}{360} - \frac{1}{2} R\sin\theta R\cos\theta]$$

$$= D_T(W_T + I) - R^2[\frac{\theta\pi}{360} - \frac{\sin 2\theta}{4}]$$

or

$$A_R = D_T(W_T + R\sin\theta) - \frac{R^2}{4}[\frac{\theta\pi}{90} - \sin 2\pi] \qquad \text{(C-1)}$$

Thus the hydraulic diameter becomes

$$D_e = \frac{4D_T(W_T + R\sin\theta) - R^2[\frac{\theta\pi}{90} - \sin 2\pi]}{\frac{2\pi R\theta}{360} + R\sin\theta + W_T + D_T}$$

where

$$\theta = \cos^{-1}(1 - \frac{D_T}{R})$$

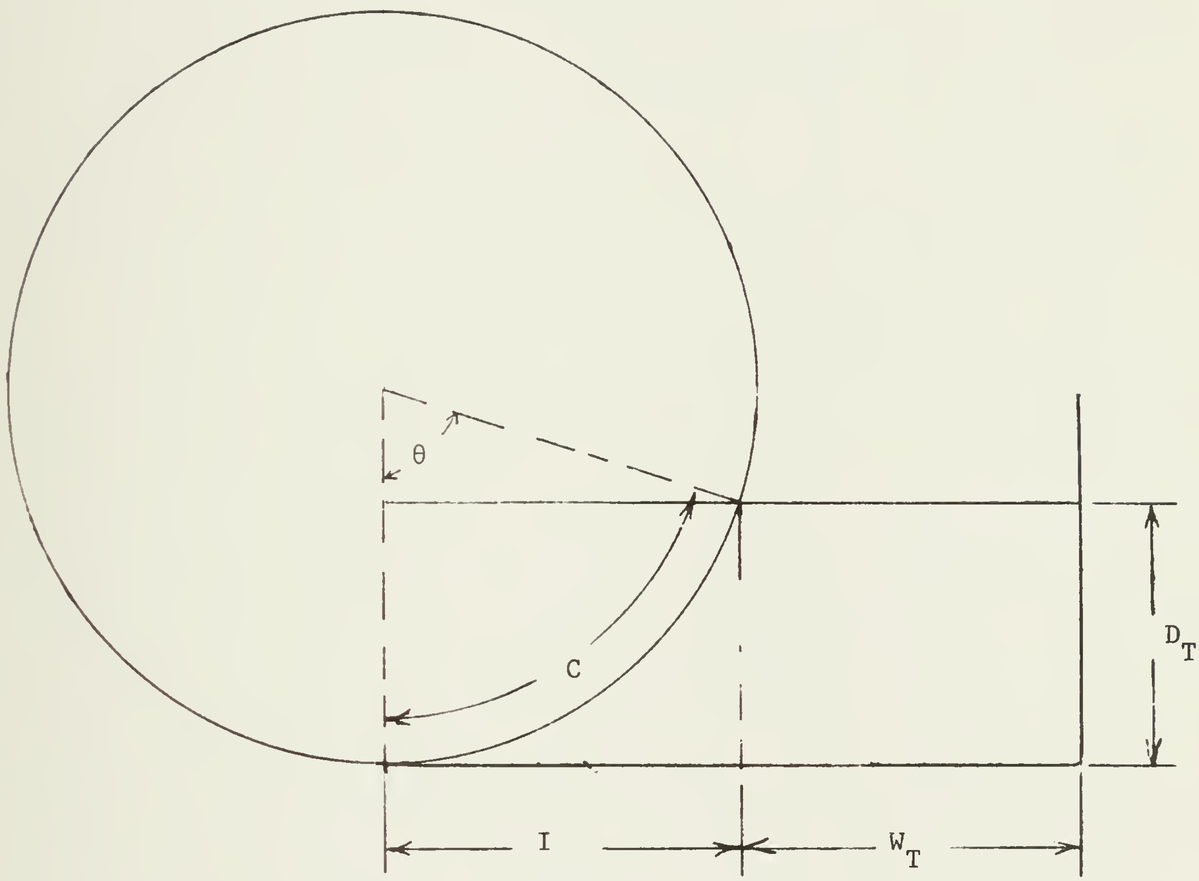

Figure C-1 Geometry of the Trough Used to Calculate the Hydraulic Diameter

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